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SURFACE SHIP FUEL SAVINGS WITH AN
OPTIMIZED AUTOPILOT

by

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Thesis Advisor: George J. Thaler

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It is shown that the second-order forces and moments create drift motions along the surge, sway, and yaw axes. As a consequence of this, the second-order forces and moments cause more fuel consumption than the first-order forces and moments, which create only oscillatory ship motions along this axes. So the sea state in the deterministic model is represented by the first-order and second-order forces and moments.

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Surface Ship Fuel Savings
With An
Optimized Autopilot

by

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ABSTRACT

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It is shown that the second-order forces and moments create drift motions along the surge, sway, and yaw axes. As a consequence of this, the second-order forces and moments cause more fuel consumption than the first-order forces and moments, which create only oscillatory ship motions along these axes. So the sea state in the deterministic model is represented by the first-order and second-order forces and moments.

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I. INTRODUCTION

An overall rise in fuel prices has led to an increasing interest in the design of autopilots for ships. The purpose of the automatic steering control is to minimize propulsion losses which are caused by added drag due to steering of the ship.

Several attempts have been made to define a measure of steering efficiency based on propulsion losses and proposals have been made for the design of an autopilot which minimizes the propulsion losses. As a performance criteria, added resistance due to steering suggested by Norrbin, N. H. was used in most of the studies because it is convenient for ship board use.

Many researchers indicate that a carefully designed controller could save from one to two percent of fuel by minimizing the propulsion losses which are caused by added drag due to steering of the ship . No analytical evaluation of propulsion losses due to steering in a sea state has been made.

The goal of this thesis is to study fuel savings using various kinds of controllers, and especially to compare an Adaptive Automatic Optimal Controller with a PID controller to see the difference between them. The optimal gain parameters of the Adaptive Automatic Optimal Controller are provided by LTJG. Cetin Diken [Ref. 1].

To study the optimization problem, models of both the ship and the operating environment are required. Chapter 2 addresses what type of computer model can be used to represent the ship.

The ship's nonlinear equations of motion were needed to simulate the ship in the computer program. Chapter 3 addresses the Mariner Class ship nonlinear equations of motion.

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Chapter 4 addresses the problems of testing the ship simulation model in calm water, of expressing the forces and moments due to sea state, and the effects of sea state on ship's behaviour.

Chapter 5 addresses the derivation of the cost function which represents the added drag due to steering.

What is an adequate way to represent the fuel consumption ratio of added drag due to steering? Chapter 6 addresses the fuel consumption ratio.

Ship dynamics change with operating conditions such as ship speed, encounter angle, encounter frequency, and sea state. Chapter 7 presents the fuel consumption ratio of added drag due to steering for PID and an Automatic Optimal Controller, and the difference between them.

Conclusions were drawn from simulation results, and are presented in Chapter 8. This chapter also recommends topics for future studies.

II. DESCRIPTION OF THE COMPUTER MODEL

Before a specific controller can be designed, a realistic model for the ship dynamics must be found to enable proper simulation of the ship movements in response to control signals.

The model which best represents ship-steering dynamics is a Taylor's series expansion of the force and moment relationship around a selected steady-state operating point. The resulting equations are commonly known as the equations of motion.[Refs. 2, 3, 4, 5]

A computer program was developed in order to provide a computer simulation for the ship, using experimentally measured hydrodynamic coefficients for the Mariner Class Ship [Refs. 2, 3, 6, 7]. Figure 2.1 shows the block diagram. The computer program is shown in Appendix C [Refs. 8, 9, 10, 11].

The function minimization subroutine used was BOXPLX which was programmed by R. Hilleary. The task of BOXPLX is to find the minimum of any function. It may be subjected to explicit constraints of the variable or implicit constraints on functions of the variables. [Refs. 11]

The sea disturbance is found as first and second-order forces and moments by a sea state program which has been written by J. Cass [Ref. 9].

D type or PID controller was used in the computer simulation program. PID controlller is shown in Appendix A. The D type controller was described by:

$$\delta = \left[\frac{K(1+T_z s)}{(1+T_p s)} + \frac{1}{T_I s} \right] \psi_e$$

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where $K, T_z, T_p,$ and T_I are optimal parameters which were found by the minimization subroutine, BOXPLX. The definitions of symbols used in equations and figures are given in Appendix D.
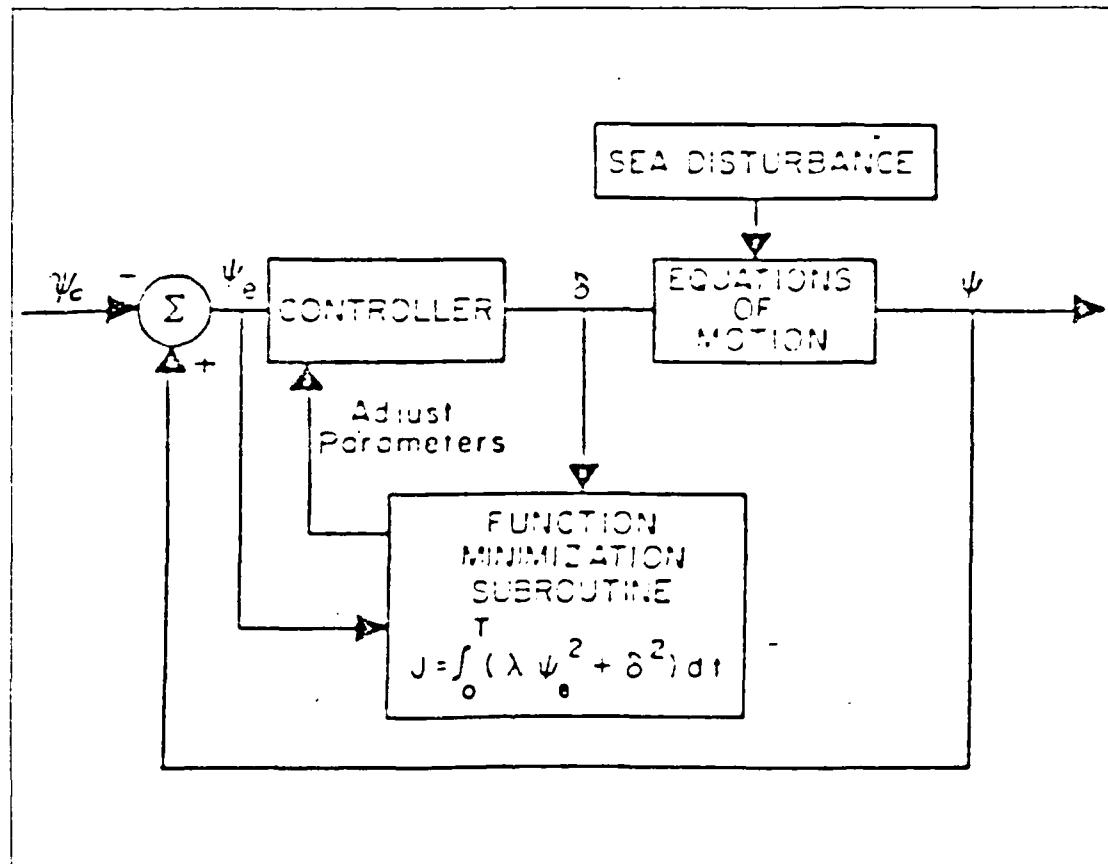
Figure 2.1   Block Diagram of Ship and Control System.

III. NONLINEAR EQUATIONS OF MOTION

Linear theory is useful for analyzing the influence of ship features on controls-fixed stability as well as on the turning ability of stable ships in the linear range [Refs. 2, 3, 4, 5, 12, 13]. However, it fails to predict accurately the characteristics of the tight maneuvers that most ships are capable of performing and it cannot predict the maneuvers of unstable ships.

Nonlinear equations of motion are suitable for predicting tight maneuvers and also suitable for computer programming. The nonlinear equations of motion based on a Taylor series expansion of forces and moments including terms up to the third order have been developed by Abkowitz [Refs. 5, 12], and Strom-Tejsen [Ref. 3]. Accuracy is not improved by including terms higher than third order. The development of these nonlinear equations is based on a restatement of linear equations to include rudder angle. Equations X, Y, and N are functions of u, v, r, u, v, r, and δ.

Combining the nonlinear Taylor series expansion of forces and moments terms up to third order with the dynamic response terms of the X, Y, and N equations, the nonlinear equations of motion are shown in Appendix B. The hydrodynamic coefficients of the equations are determined from experimental data obtained from captive model tests. They are given in Appendix B [Ref. 7]. The terms not included in tables are negligible.

IV. SHIP'S BEHAVIOUR IN CALM WATER AND SEA STATES

A. CALM WATER

A simulation program in Appendix C was run for turning and zig-zag maneuvers to observe u, v, r, and ψ using the ship's nonlinear equations of motion and Mariner Class Ship coefficients.

It was observed that the rudder angle changes the ship's course. The ship's speed decreased while turning. The absolute value of v and r increased and after a while they reached steady-state values, since the ordered rudder angle is constant. The larger the rudder angle, the greater the decrease in speed, and the faster the ship turns.

Time responses of r, v, and u are shown in Figure 4.1, 4.2, 4.3, and X vs. Y is shown in Figure 4.4 for the turning maneuver with δ = 25 degree. δ, ψ, and u are shown in Figure 4.5 and Figure 4.6 for the zig-zag maneuver. As is seen from the figures, a suitable and sufficiently accurate ship computer model was defined.

B. SEA STATE

To observe the ship's behavior in a sea state, disturbance forces and moments are needed. They depend on sea state, ship speed, encounter angle, and encounter frequency. The added mass and added inertia are functions of encounter frequency and sea state.

A regular sea model was used as the sea representation: the wave crests assumed to be straight, infinitely long, parallel, and equally spaced with constant wave height [Refs. 2, 9, 14, 15].

The forces exerted by the regular sea can be represented as the sum of two components, called first and second-order forces [Refs. 2, 6, 9, 14, 16, 17, 18, 19, 20].

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The first-order exciting forces are a zero mean process. The second-order forces are a nonzero mean process, i.e. , they have essentially constant values. In regard to maneuvering, the first-order forces are not the primary mover of the ship. The reason for this is that the high frequencies of the first-order forces are higher than the ship can readily respond to. The second-order forces cause the large excursions that must be manually controlled, while the first-order forces cause only a ripple on the ship surface trajectory [Ref. 6].

The sea also induces moments on the hull of a surface ship. These can be represented by first and second-order moments which possess the same basic characteristics as the first and second-order forces, and contribute to angular motion in a similar way.

The first-order forces exerted by the regular sea have the form:

$$F = W_a R_i Cos(\omega_e t + \phi_i) \qquad (eqn\ 4.1)$$

where

$$\omega_e = (\omega_{wave}^2 /g) [(g/\omega_{wave}) - u Cos\beta)]$$

The second-order forces FXX, FYY, and moment MZZ have constant components and periodic components at twice the encounter frequency [Refs. 6, 9, 20]. The second-order forces FXX, FYY, and moment MZZ were assumed to approximate the constant drifting forces and moments calculated by using Sea State Program of James L. Cass [Ref. 9] for the Mariner Class ship in regular waves. This approximation was used because these were the data available at the time. The sinusoidal part of the the second-order forces and moments (frequency of 2 ω_e) was neglected at this point of the work

[Ref. 6]. The high frequencies of the second-order forces and moments are also higher than the ship can respond to. They do not effect the ship surface trajectory.

The exciting forces R_i, and second-order forces for different encounter frequencies and encounter angle were obtained from the sea state program [Ref. 9]. Data input to the sea state program for Mariner Class Ship is shown in [Refs. 1, 10].

To see the ship's behaviour in sea state, the simulation program was run for ship speed 15 Knots, encounter angle 120 degree, encounter frequency 0.64 radian/second, and sea state 8. First and second-order forces and moments were added into the surge, sway, and yaw equations that were used in the simulation program in Appendix C. Time responses of u, ψ and δ are presented in Figure 4.7, 4.8, and 4.9. In these simulations, optimal gain parameters found by the BOXPLX subroutine were used with autopilot.

The first-order forces and moments are sinusoidal, therefore they create only oscillatory ship motion along the surge, sway, and yaw axes. The second-order forces and moments are constant, thus creating drift motions along these axes.

It is apparent that as the encounter frequency increases, the effect of the first-order forces on ship motions decreases because of the high ship inertia (acting as a low pass filter) [Ref. 10, 21, 22]. An increasing encounter frequency means a decreasing wavelength-to-ship length ratio. The constant value of the second-order forces increases when the wavelength-to-ship length ratio decreases. Regarding the theory, forces and moments are significant for short wavelengths. [Refs. 9, 17]

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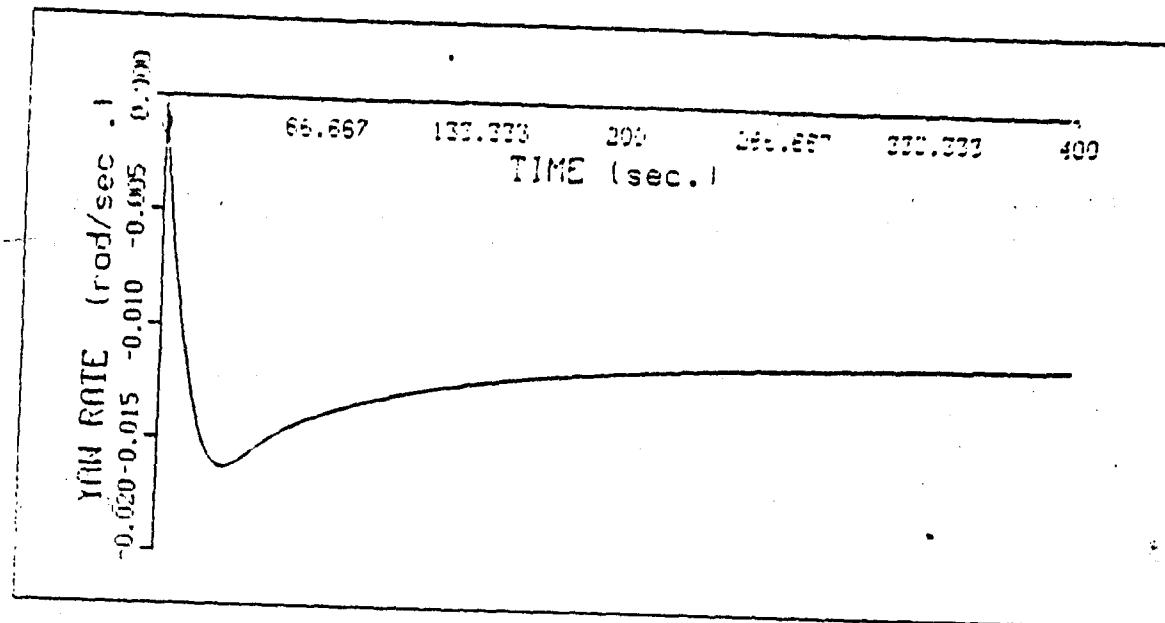
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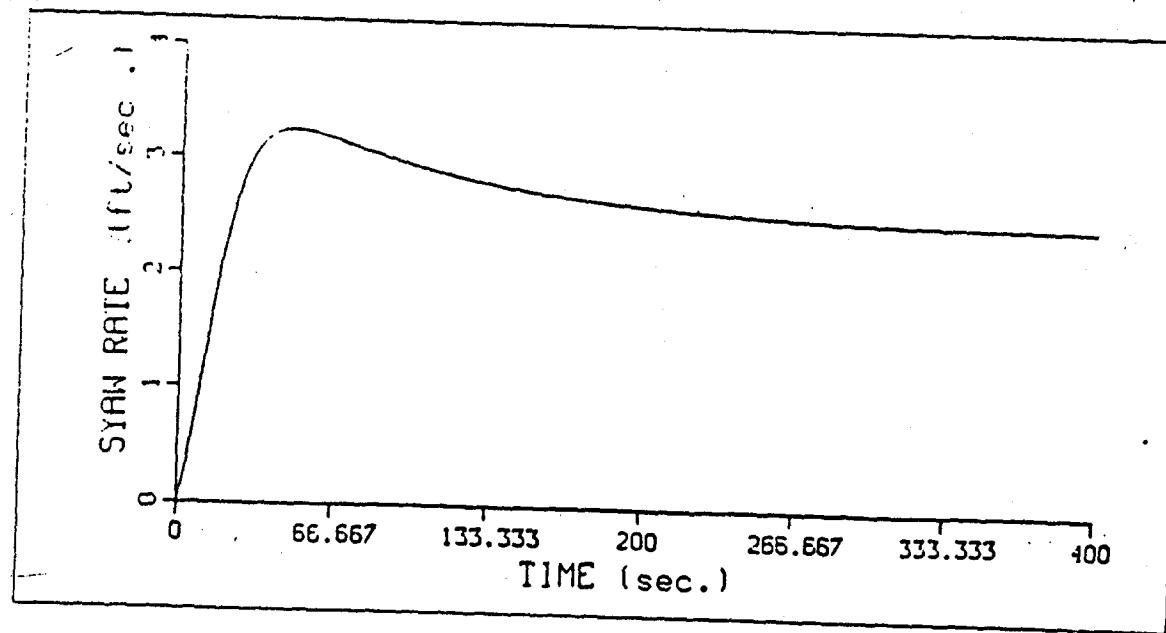
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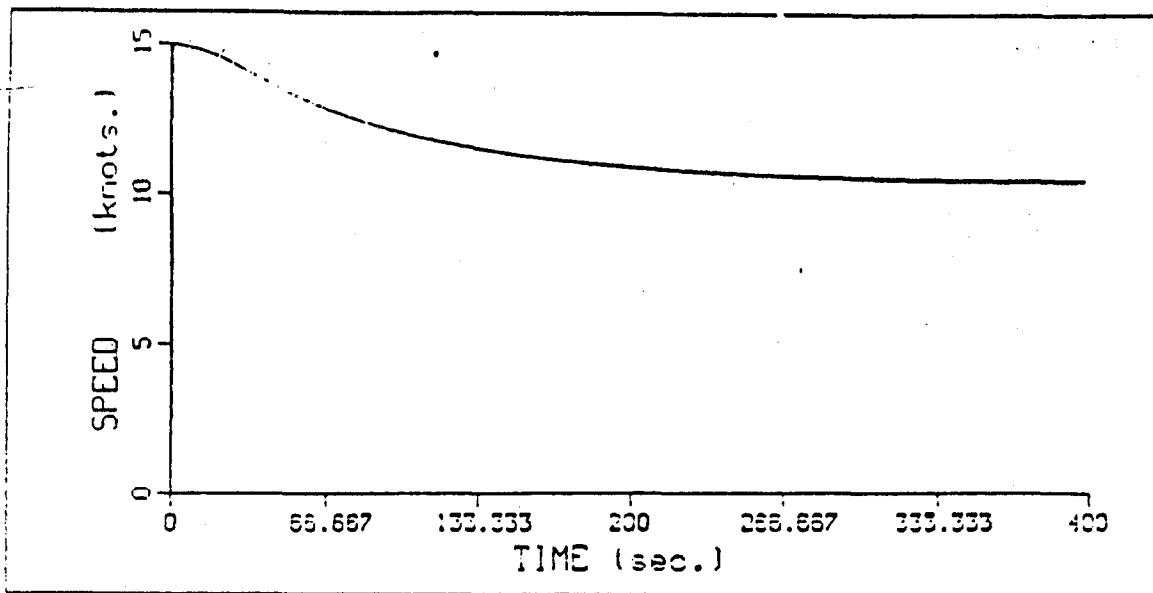
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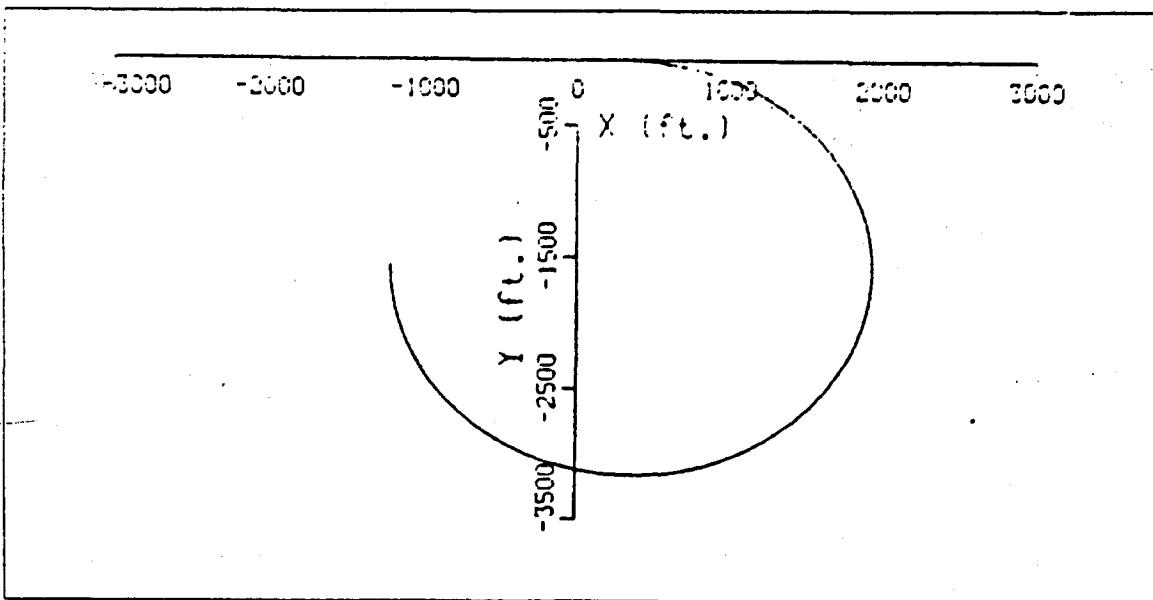
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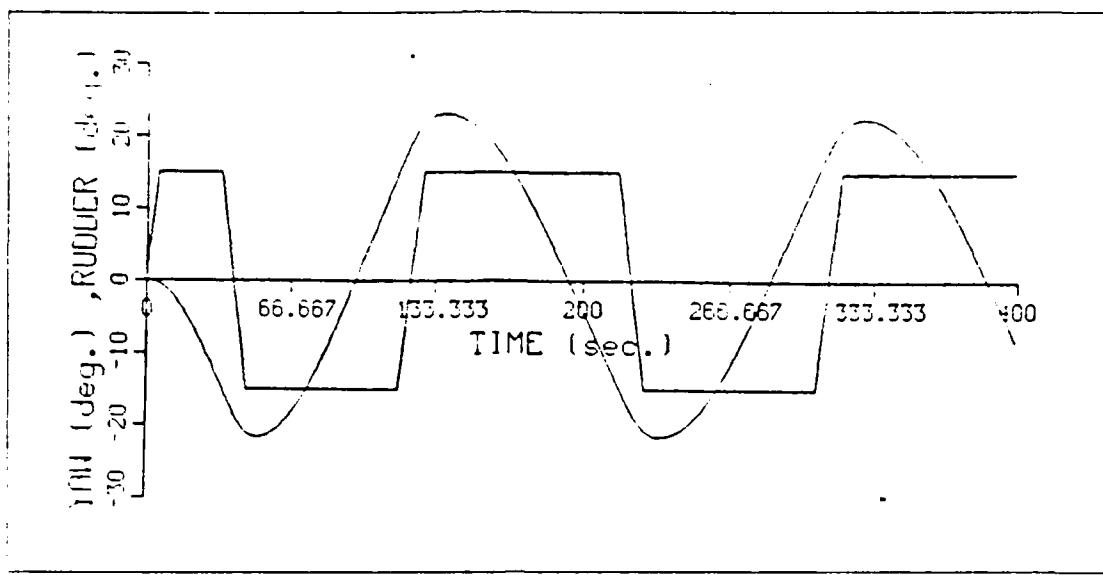
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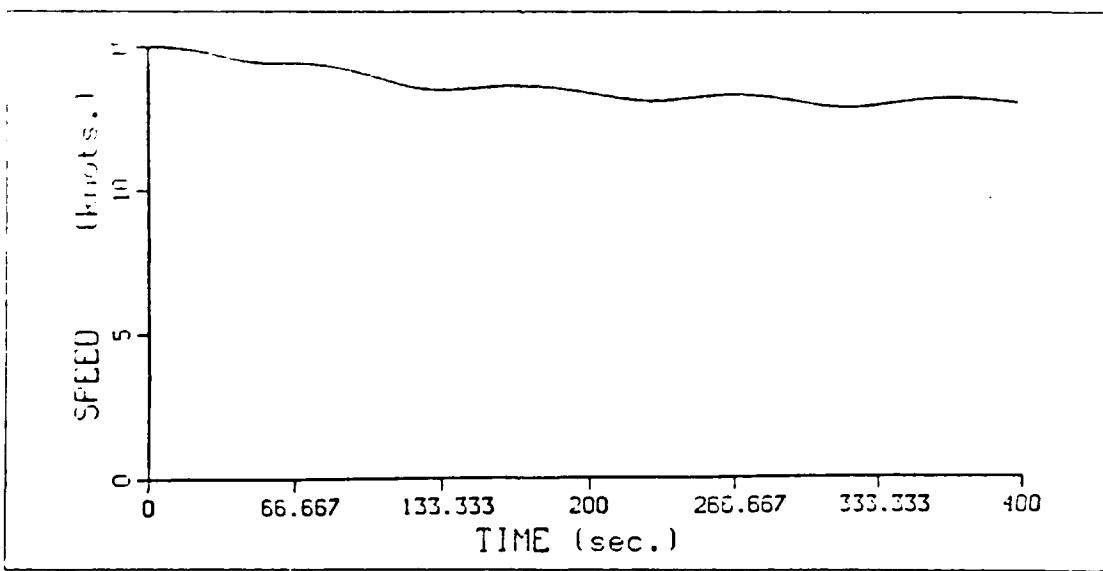
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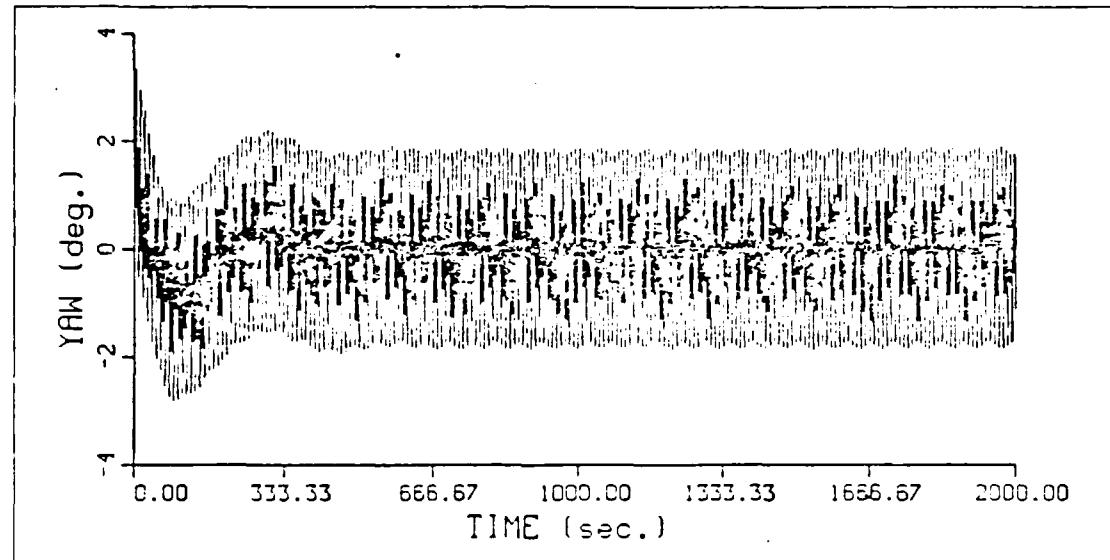
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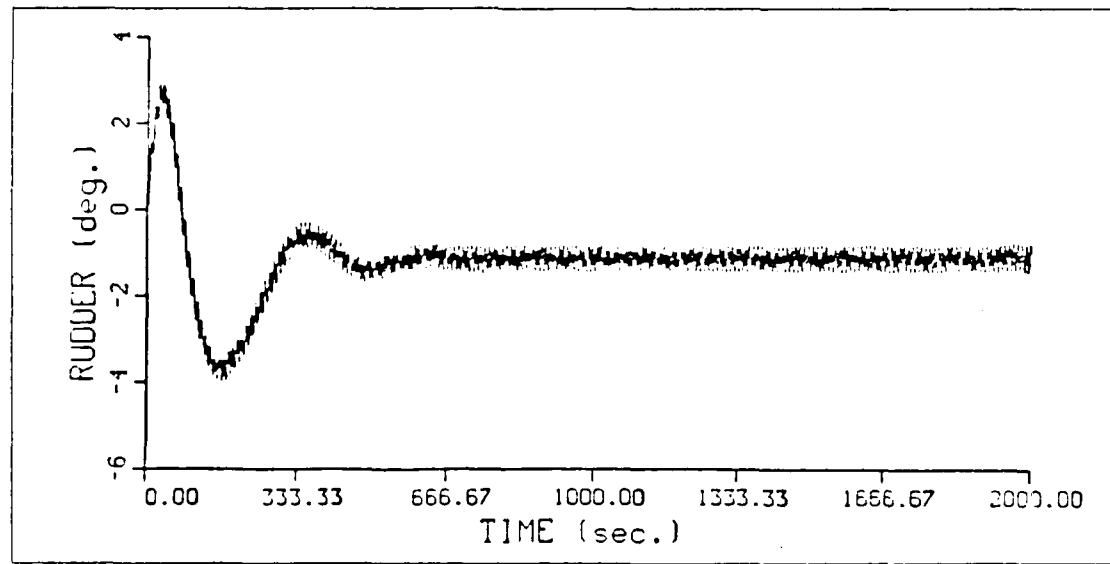
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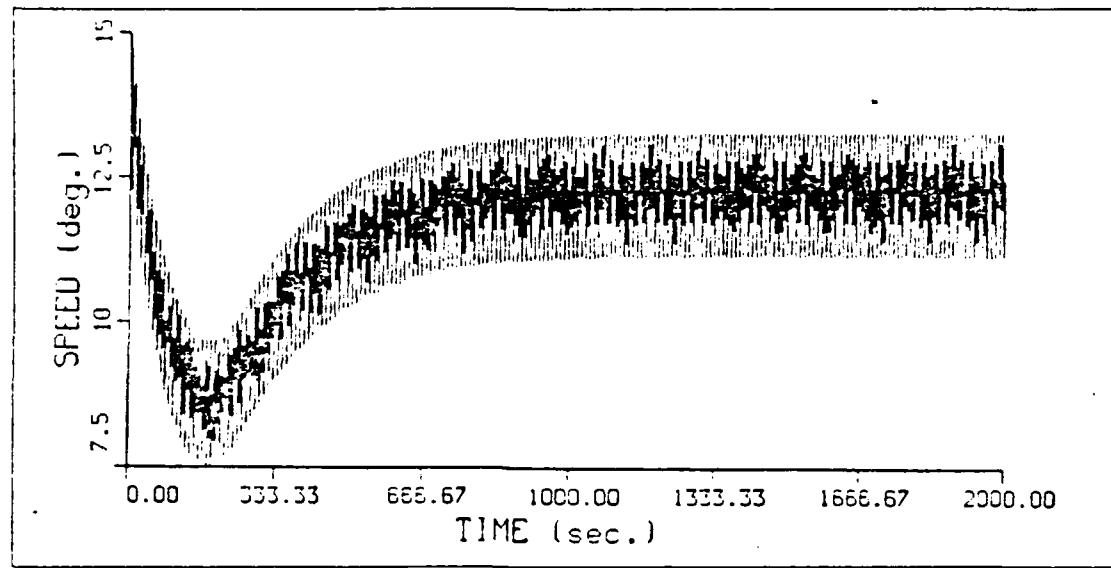
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V. COST FUNCTION

It is accepted that the performance objective of the system in open-sea course keeping is " minimum added resistance due to steering " . For Routine predictions of the fair-weather trial speeds for ships, the increase of resistance due to steering is often taken to be one percent of the towline resistance. Although this may offer little for improvement, when it is based on a fuel cost 0.5 percent of 10 million dollars a year still equals 50000 dollars. Moreover, in moderate to bad weather the benefits of good steering rapidly become more evident. It is difficult to measure the changes of rate of fuel consumption due to different autopilots. A number of alternative cost functions have been suggested to describe the rate of fuel consumption [Refs. 23, 24, 25, 26, 27, 28].

When deriving a cost function for the autopilot, a requirement is that it must be convenient for ship board use. The cost function that has been commonly used in recent years is:

$$J = \lim_{T \to \infty} (1/2T) \int_0^T (\lambda \psi^2 + \delta^2) \, dt \qquad \text{(eqn 5.1)}$$

This is an approximate form of the exact cost function.

A. EXACT FORM

The performance criterion which characterizes the propulsion losses due to steering may be shown to be that derived from excess power consumption per unit distance caused by the added resistance due to steering. The added resistance due to steering can be related to the surge equation where the total instantaneous surge relevant to steering is [Ref. 29]:

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$$\Delta X = (1/2)X_{vv}v^2 + (1/2)X_{\delta\delta}\delta^2 - (X_{vr}+m)vr$$

$$+ (1/2)(X_{rr}+mX_G)r^2 + X_{v\delta}v\delta \qquad \text{(eqn 5.2)}$$

This may be used to determine the energy losses related to steering in both calm water and waves. It is not convenient for ship board use, but may be used when finding fuel cost and comparing two controllers in fuel consumption.

Then the exact form of the performance criteria is:

$$J = \lim_{T\to\infty} (1/2T) \int_0^T [(1/2)X_{vv}v^2 + (1/2)X_{\delta\delta}\delta^2 + (X_{vr}+m)vr$$

$$+ (1/2)(X_{rr}+mX_G)r^2 + X_{v\delta}v\delta]dt \quad \text{(eqn 5.3)}$$

The lower limit is written as zero, but while using a simulation program, the cost function will be calculated after t_0, the transient response time of the system.

B. APPROXIMATE FORM

Accurate knowledge of the nonlinear coefficients in the exact form of the performance criterion is required for accurate results. In addition the criterion itself suffers from the disadvantage that the sway velocity is not measurable in practice.

How can the approximate form which is suitable for ship board be found [Refs. 10, 23, 24]?

Since the sway velocity of the ship is small, the term including the square of the sway velocity can be neglected. It is also seen that $X_{v\delta}v\delta$ and $(1/2)(X_{rr}+mX_G)r^2$ terms are small compared to others [Ref. 6]. After these assumptions,

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the cost function for the Mariner Class Ship can be written as:

$$J = \lim_{T \to \infty} (1/2T) \int_0^T [(1/2)X_{\delta\delta}\delta^2 + (X_{vr}+m)vr]dt \qquad \text{(eqn 5.4)}$$

v and r are difficult to measure on ship board. vr can be defined in more convenient form for ship board use.

Yawing and swaying of the vessel is assumed to stem from either self oscillations due to the steering system or the forced oscillations due to waves. The ship motions resulting from oscillatory seaway disturbance are oscillatory. Although these oscillations do not exactly follow a sinusoidal pattern, they may be approximated as regular yawing of simple periodic form. Then yaw rate, sway velocity, and rudder angle can be represented as:

$$r = r_a \sin(\omega t + \phi_r) \quad .$$

$$v = v_a \sin(\omega t + \phi_v) \qquad \text{(eqn 5.5)}$$

$$\delta = \delta_a \sin(\omega t + \phi_\delta)$$

Therefore:

$$J = (1/2)X_{\delta\delta}\delta^2 \sin^2(\omega t + \phi_\delta)$$

$$+ (X_{vr}+m)(v_a r_a/2) \cos(\phi_v - \phi_r) \qquad \text{(eqn 5.6)}$$

It is assumed that during low frequency oscillations $\cos(\phi_v - \phi_r) = -1$, i.e. , yaw rate and drift angle are in phase with one another. And writing $\beta = v/U$ as drift rate and assuming small amplitude oscillations around the pivot point p then, from Figure 5.1:

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$$\beta = (OP/R) = (OP/L)*(L/R) \qquad \text{(eqn 5.7)}$$

where

$$R = (U/r)$$

Therefore:

$$vr = ru\beta = ru(OP/L)(L/R) = r^2 \, OP \qquad \text{(eqn 5.8)}$$

where

$$r = \dot{\psi} = \omega\psi$$

Then the expression for the cost function becomes:

$$J = \lim_{T \to \infty} (1/2T) \int_0^T [(1/2)X_{\delta\delta}\delta^2 + (X_{vr}+m) \, OP \, \omega^2\psi^2] dt \qquad \text{(eqn 5.9)}$$

Then:

$$J = \lim_{T \to \infty} (1/2T) \int_0^T (\lambda\psi^2+\delta^2) dt \qquad \text{(eqn 5.10)}$$

where

$$\lambda = \frac{(X_{vr}+m) \, OP\omega^2}{(1/2)X_{\delta\delta}}$$

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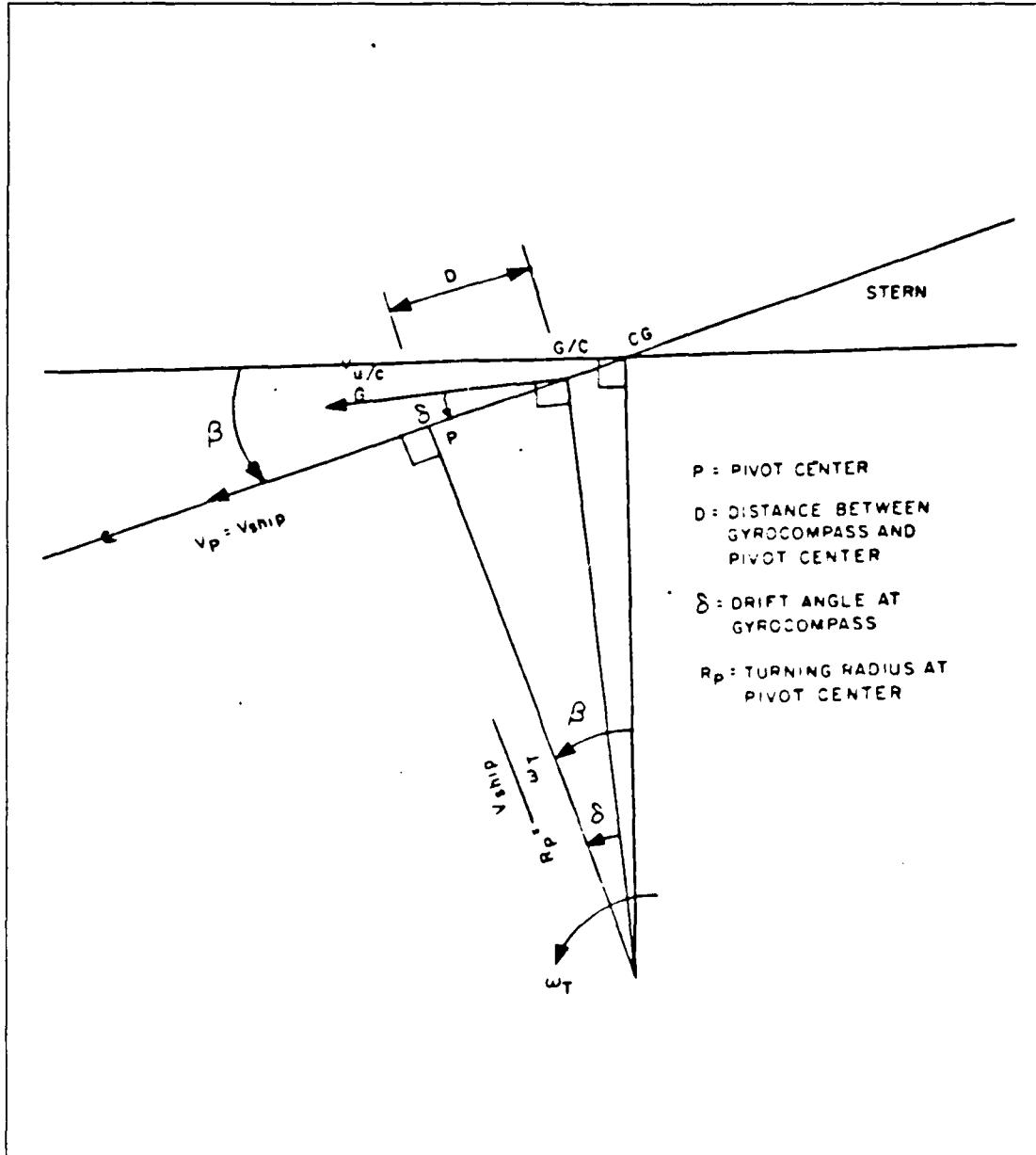
Figure 5.1   Geometry of Ship Turning.

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VI. FUEL CONSUMPTION

The accuracy of the ship's path is irrelevant on the open sea. The important thing is minimizing the propulsion power losses. These propulsion power losses can be caused by added drag due to steering and external forces and moments due to sea waves and wind acting on the ship.

Because ψ_e and δ values are easily measured, the optimum control parameters can be found by using the approximate form of the performance criterion. However, this does not provide evaluation of fuel consumption, nor does it permit comparison of different controller designs.

Since ship operation may try to maintain constant power at the engine, or may attempt to maintain constant speed, autopilot design might be required for either condition of operation.

A. CONSTANT ENGINE POWER

In calm water, the ship is allowed to proceed forward at a steady speed, and is free from lateral disturbance while undertaking part of a voyage. In this case the ship should take an elapsed time t, to cover a distance s, and speed u. In reality, the ship does encounter lateral disturbances which causes it to yaw at angular velocity r, sway sideways with velocity v, and rudder resistance is increased while keeping the course. This causes the ship to travel a longer path (Δs) than necessary, a longer time (Δt) is required than for the undisturbed case, and there is speed loss (Δu).

The engine uses a certain amount of fuel in a given time:

$$P = W_{net} \cdot m \qquad \qquad \text{(eqn 6.1)}$$

$$\text{Fuel Consumption} = P \cdot t \qquad \text{(eqn 6.2)}$$

In the disturbed case:

$$\text{Fuel Consumption} = P \cdot t' \qquad \text{(eqn 6.3)}$$

where

$$t' = t + \Delta t$$

The ship path can be simulated with the computer simulation program in Appendix D. And the time to travel the same distance along the x-axes can be determined for calm water and for the sea state case.

In fact, to find how much fuel is consumed, the engine specifications must be known. But, when different controllers are compared to determine the percent fuel savings in the same situations, there is no need for engine specifications.

$$\text{\% Fuel Consumption Ratio} = \frac{P\,t'_2 - P\,t'_1}{P\,t'_1} \times 100 \qquad \text{(eqn 6.4)}$$

where

$P \bullet t'_1$ = Fuel consumption at t'_1, elapsed time to travel the given distance along the x-axes while using controller #1(assume as an automatic optimal controller)

$P \bullet t'_2$ = Fuel consumption at t'_2, elapsed time to travel the given distance along x-axes while using controller #2(assume as a PID controller)

then:

$$\text{\% Fuel Consumption Ratio} = \frac{t'_2 - t'_1}{t'_1} \times 100 \qquad \text{(eqn 6.5)}$$

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The sea state forces and moments, and added drag due to steering affect the elapsed time to travel the given distance along the x-axes. This ratio gives the fuel consumption ratio including all sea state forces, moments, and added drag due to steering. But, the effect of added drag due to steering for different controllers is needed to compare fuel consumption.

The other disadvantage is that it can be used only for small sea state disturbances, since the ship speed decreases too much in high sea state. It may be even sufficient to drive the ship backwards [Ref. 6]. The vital effect of the decreasing speed is that this changes the values of the hydrodynamic coefficients of the nonlinear equations given in Table 2, 3, and 4 in Appendix B. The big speed differences from steady-state speed, 15 Knots, makes the nonlinear coefficient values incorrect to represent the ship-steering dynamics.

As a result; for small disturbances, i.e. , when the representation of ship-steering dynamics does not change, then this method may be used. But, for large sea disturbances, the ship speed must not be allowed to decrease much to affect the representation of ship-steering dynamics in the simulation program.

B. CONSTANT SPEED

Using Table 6.1 [Ref. 6], it is seen that the added drag force due to increasing RPM is a linear function. The linear equation " added force vs. RPM " is found, and used in the simulation program to keep the speed constant.

While attempting to maintain speed constant, it is more difficult to find the fuel consumption ratio, since P is varying. P cannot be factored out, as in Eqn. 6.5.

How can a method to find the fuel consumption ratio without using engine specifications be found?

The added resistance due to steering is:

$$\Delta X = (1/2)X_{vv}v^2 + (1/2)(X_{rr}+mX_G)r^2 + (1/2)X_{\delta\delta}\delta^2$$

$$+ (X_{vr}+m)vr + X_{v\delta}v\delta \qquad \text{(eqn 6.6)}$$

The total surge equation is:

$$X_{total} = X + X_{calm} \qquad \text{(eqn 6.7)}$$

where

$$X_{calm} = T (1 - t)$$

The fuel consumption ratio of added resistance due to steering to the total surge equation, assuming constant overall propulsive efficiency, [Refs. 2, 25] is:

$$\text{\% Fuel Consumption Ratio} = \frac{\Delta x}{X_{total}} \times 100 \qquad \text{(eqn 6.8)}$$

Using this ratio, it is possible to compare the fuel consumption of the Automatic Optimal Controller and with that of the PID. To find exact fuel consumption, engine specifications are still needed. This ratio will be used in Chapter 7 to compare the PID with the Automatic Optimal Controller.

TABLE I
SPEED AND ADDED FORCE VS. RPM

RPM	Speed (Knots)	Added Force (LB)
55.5	7.0	13320
56.8	8.0	16870
58.1	9.0	21070
59.6	10.0	25710
61.2	11.0	31400
62.8	12.0	37780
64.6	13.0	45500
66.5	14.0	53960
68.6	15.0	63420

VII. COMPARISON OF PID AND AUTOMATIC OPTIMAL CONTROLLER

The Automatic Optimal Controller offers the potential for minimization of propulsion losses due to steering in the open sea, and for removal of operator judgement from steering control of the ship. The optimal parameters which were found by using the BOXPLX subroutine in the Optimal Gain Program were shown in Table II [Ref. 1]. The approximate form of the cost function was used in the BOXPLX subroutine.

Time responses of ψ and δ are shown in figure 7.1 through figure 7.10 for different encounter angles using the optimal parameter values in Table II .

No provision for automatic adaptivity to either speed, load or seaway exists in the Universal Gyropilot, PID. Some adjustments of control parameters are possible, however, through an operator interface. Details of such adjustments, and the structure and parameter values of a PID controller are contained in Appendix A.

Although a PID controller can come close to emulating this performance under specified internal and operator settings, the operator has no means to judge steering performance other than by observation of the course recorder. The normal tendency of the operator is to change the external controls available to him to bring about the reduction in heading error. It greatly increases the propulsion losses due to steering.

The goal in this chapter is to compare an Automatic Optimal Controller with a well tuned PID controller in fuel savings. Now the question is how the Well-Tuned PID can be defined. There are two ways to answer this question. One way is to define a well tuned PID as an Optimal PID Controller and to use the same performance criterion used in finding optimal parameters of a D Type Automatic Controller, i.e.,

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the minimization of added resistance due to steering. The optimal control parameters of a PID in different sea state, speed, encounter angle, and encounter frequencies were found by using the simulation program to compute the cost function of all combinations of internal and external control parameters, making the modifications in the simulation program shown in Appendix C. The results were shown in Table III .

Time responses of ψ and δ are shown in Figure 7.11 through 7.20 for different encounter angles using the optimal parameter values in Table III . As it is seen from the figures, the Automatic Optimal Controller and PID with optimal parameters after the transient time makes the ship yaw sinusoidally around the commanded course. To see their effects in added resistance due to steering, and in fuel savings, the comparison of them was shown in Table IV . The added resistance due to steering and fuel consumption ratio are almost the same, since the same performance criterion was used, and all internal and external controls of PID are adjusted, not only the external controls.

The PID with optimum parameters cannot be used as a well tuned PID, since the operator could not find the optimum parameters by himself to minimize the added resistance due to steering and also could not change the internal control parameters. But it is clear that if the optimum control parameters are found in the same way as for the D Type Controller, it gives the same fuel savings as an optimal D Type controller.

The other way is to approach the question as an operator. There are no means to adjust the external controls available to the operator to minimize the propulsion losses due to steering other than by observation of the course recorder. The operator can adjust the external controls to reduce the heading error, the only system output available to him. The external controls available to the operator are Weather Adjust Gain, Rudder Multiplier Gain, and Rudder Rate

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Multiplier. To reduce the heading error, the Weather Adjust Gain is set to "0", and to get the fast response of rudder, the Rudder Multiplier Gain is at "3", and the Rudder rate Multiplier setting is "1". These correspond to K_1 is "1", K_2 is "3", and the value of T_1 is half of the internal control setting.

The internal controls T_1, T_2, and T_3 are not known for the Mariner Class Type ship. These internal control parameters are found by running the simulation program used in finding optimum parameters for the PID with K_1 and K_2 set to "1", and the proportional part of the PID is omitted in the control law of PID, Eqn. A.1. The results are following:

$$T_1 = 25$$

$$T_2 = 15$$

$$T_3 = 1000$$

Time responses of ψ and δ are shown in Figure 7.21 through 7.30. As is seen in Figure 7.22, 7.24, 7.26, 7.28, and 7.30, the well tuned PID uses a larger rudder angle to reduce the heading error quickly. The rudder angle also oscillates in large magnitudes. It tries to get the ship to the command heading angle as fast as possible. It increases the cost. So consequently the fuel cost and added resistance due to steering increase. The comparison in fuel savings and in added resistance due to steering is shown in Table V . Although the optimal internal control parameters are used in a well tuned PID, there are differences in fuel consumption ratio and in added resistance due to steering, especially at quartering sea waves, i.e., 030-060 and 120-150 degrees. The Automatic Optimal Controller provides fuel savings in excess of 0.5 % over a well tuned PID when operating at the design speed at random headings in sea states.

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The internal control parameters of a well tuned PID in the deterministic model are represented by optimal values. They might not be the actual values used for a Mariner Class ship. The differences in fuel consumption ratio and in added resistance due to steering should increase if actual values are not equal to the optimal internal control parameter values used in the simulation computer program. It is also possible that the different operators can set the Rudder Multiplier Gain and Rudder Rate Multiplier in different ways. The fuel consumption ratio and added resistance due to steering also changes in relation to these values.

TABLE II

OPTIMAL PARAMETERS OF D TYPE CONTROLLER

Sea State = 8
Ship Speed = 15 Knots
Wave Frequency = 0.53

Encounter Angle	K	T_z	T_p	K_d
030	0.69	0.01	34.38	569.14
060	6.70	0.01	73.47	23.16
090	7.45	0.19	60.58	990.44
120	2.77	1.75	35.79	29.76
150	2.41	0.01	6.29	12.42

TABLE III

OPTIMAL PARAMETERS OF PID CONTROLLER

Sea State = 8
Ship Speed = 15 KNOTS
Wave Frequency = 0.53

Encounter Angle	K_1	K_2	T_1	T_2	T_3
030	0.33	1.5	10.	15.	200
060	1.	1.5	25.	10.	200
090	1.	1.	25.	15.	200
120	1.	1.	2.5	15.	200
150	1.	2.	2.5	15.	200

TABLE IV

COMPARISON OF OPTIMUM PID WITH AUTOMATIC OPTIMAL CONTROLLER

Sea State = 8
Ship Speed = 15 Knots
Wave Frequency = 0.53
Added Resistance (libre)

Encounter Angle 030 deg.

	D Type	Optimum PID	Difference
Added Resistance Due to Steering	5547	5892	345
Fuel Consumption Ratio (%)	6.106	6.552	0.446

Encounter Angle 060 deg.

	D Type	Optimum PID	Difference
Added Resistance Due to Steering	52422	52734	312
Fuel Consumption Ratio (%)	33.947	34.080	0.133

Encounter Angle 090 deg.

	D Type	Optimum PID	Difference
Added Resistance Due to Steering	108930	109250	320
Fuel Consumption Ratio (%)	59.749	59.928	0.179

Encounter Angle 120 deg.

	D Type	Optimum PID	Difference
Added Resistance Due to Steering	51377	51857	210
Fuel Consumption Ratio (%)	28.179	28.442	0.179

Encounter Angle 150 deg.

	D Type	Optimum PID	Difference
Added Resistance Due to Steering	14756	15221	465
Fuel Consumption Ratio (%)	8.052	8.307	0.255

TABLE V

COMPARISON OF WELL TUNED PID WITH AUTOMATIC OPTIMAL CONTROLLER

Sea State = 8
Ship Speed = 15 Knots
Wave Frequency = 0.53

Encounter Angle 030 deg.

	D Type Cont.	Well Tuned PID	Difference
Added Resistance Due to Steering	5547	7178	1631
Fuel Consumption Ratio (%)	6.106	7.761	1.655

Encounter Angle 060 deg.

	D Type Cont.	Well Tuned PID	Difference
Added Resistance Due to Steering	52422	53894	1108
Fuel Consumption Ratio (%)	33.947	34.340	0.393

Encounter Angle 090 deg.

	D Type Cont.	Well Tuned PID	Difference
Added Resistance Due to Steering	108930	109267	337
Fuel Consumption Ratio (%)	59.749	59.934	0.185

Encounter Angle 120 deg.

	D Type Cont.	Well Tuned PID	Difference
Added Resistance Due to Steering	51377	53561	2184
Fuel Consumption Ratio (%)	28.179	29.375	1.196

Encounter Angle 150 deg.

	D Type Cont.	Well Tuned PID	Difference
Added Resistance Due to Steering	14756	15747	991
Fuel Consumption Ratio (%)	8.052	8.593	0.541
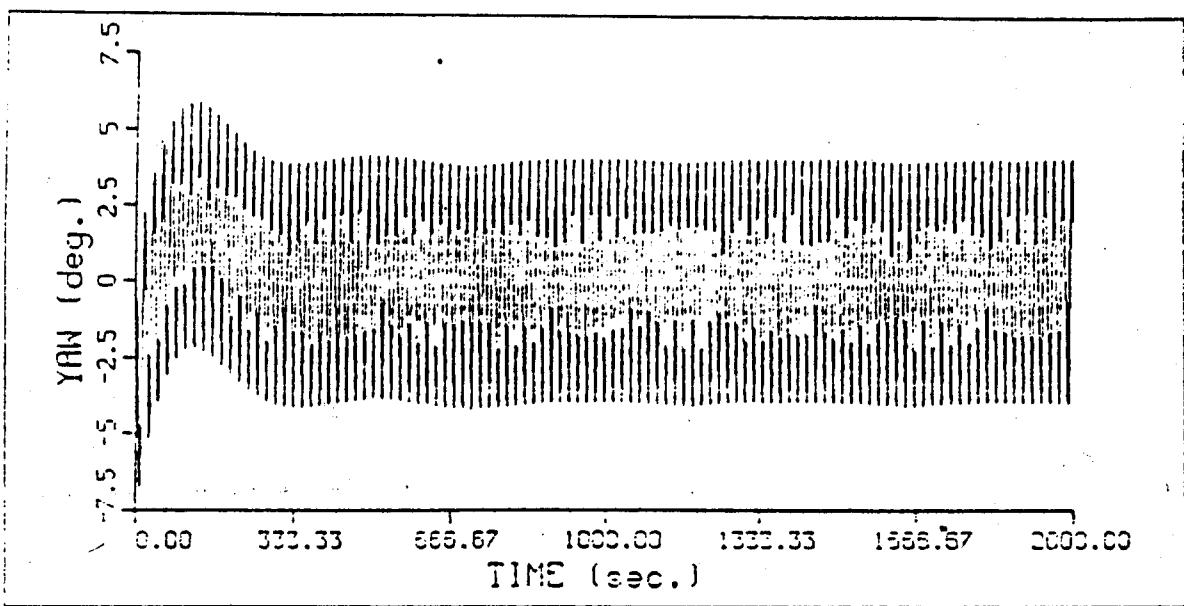
Figure 7.1   S. State=8, D Type Cont., Speed=15 Knots
Encounter Angle=030, Encounter Frequency=0.33
Yaw Degree vs. Time.
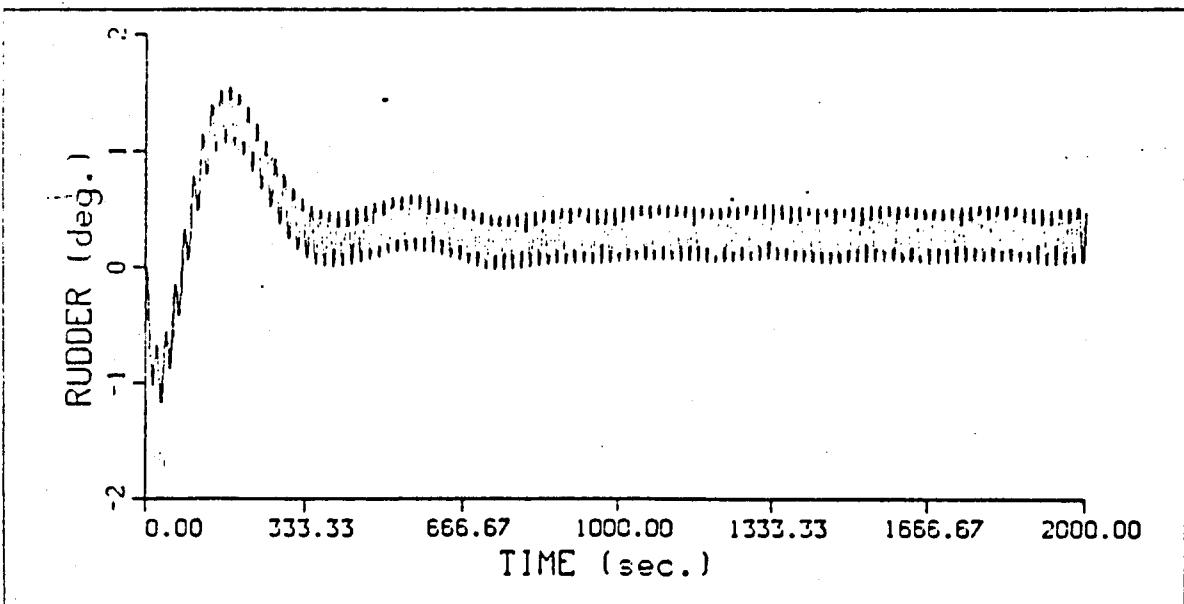
Figure 7.2   S. State=8, D Type Cont., Speed=15 Knots
Encounter Angle=030, Encounter Frequency=0.33
Rudder vs. Time.
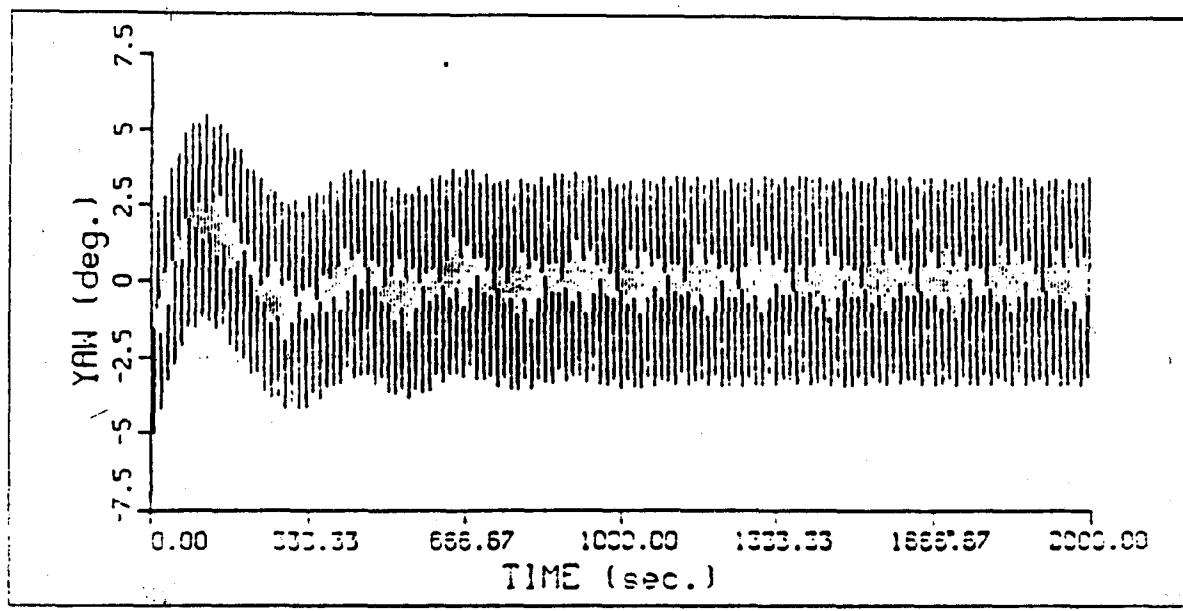
Figure 7.3   S. State=8, D Type Cont., Speed=15 Knots
Encounter Angle=060, Encounter Frequency=0.43
Yaw Degree vs. Time.
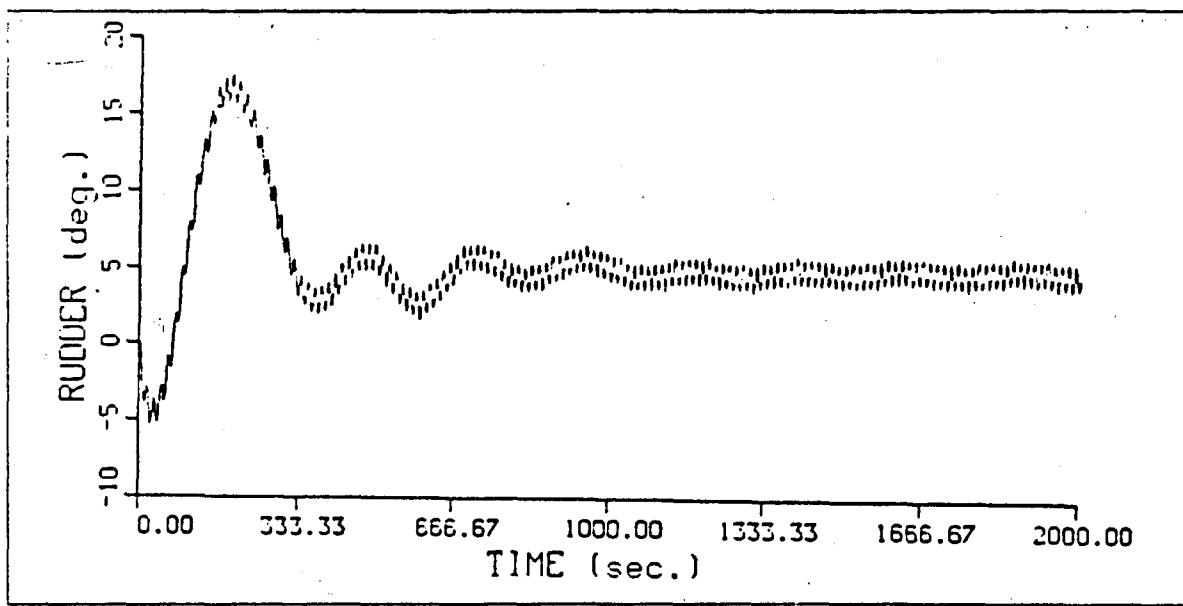
Figure 7.4   S. State=8, D Type Cont., Speed=15 Knots
Encounter Angle=060, Encounter Frequency=0.43
Rudder vs. Time.
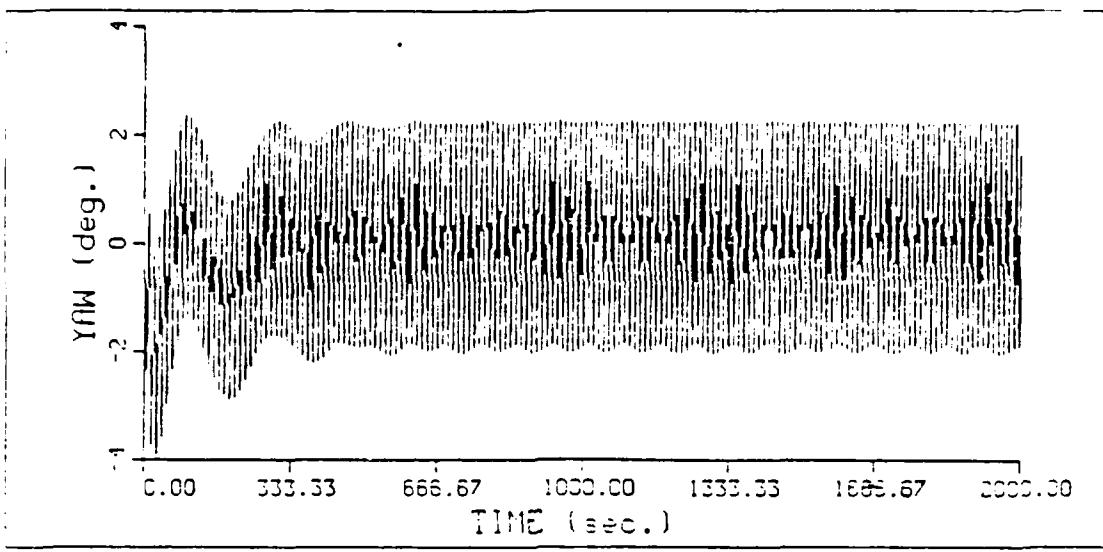
Figure 7.5   S. State=8, D Type Cont., Speed=15 Knots
Encounter Angle=090, Encounter Frequency=0.53
Yaw Degree vs. Time.
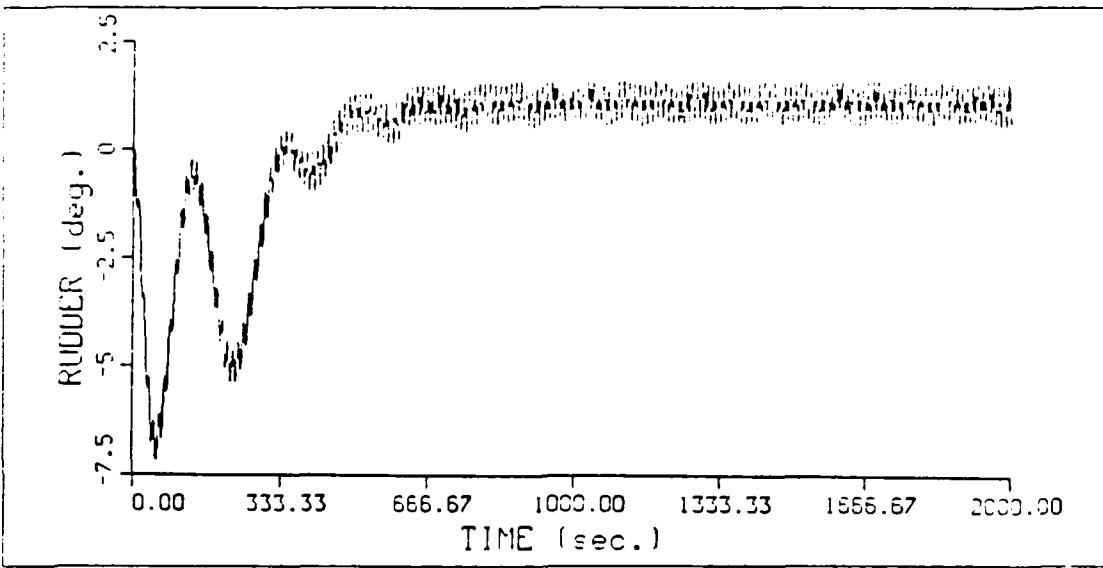
Figure 7.6   S. State=8, D Type Cont., Speed=15 Knots
Encounter Angle=090, Encounter Frequency=0.53
Rudder vs. Time.

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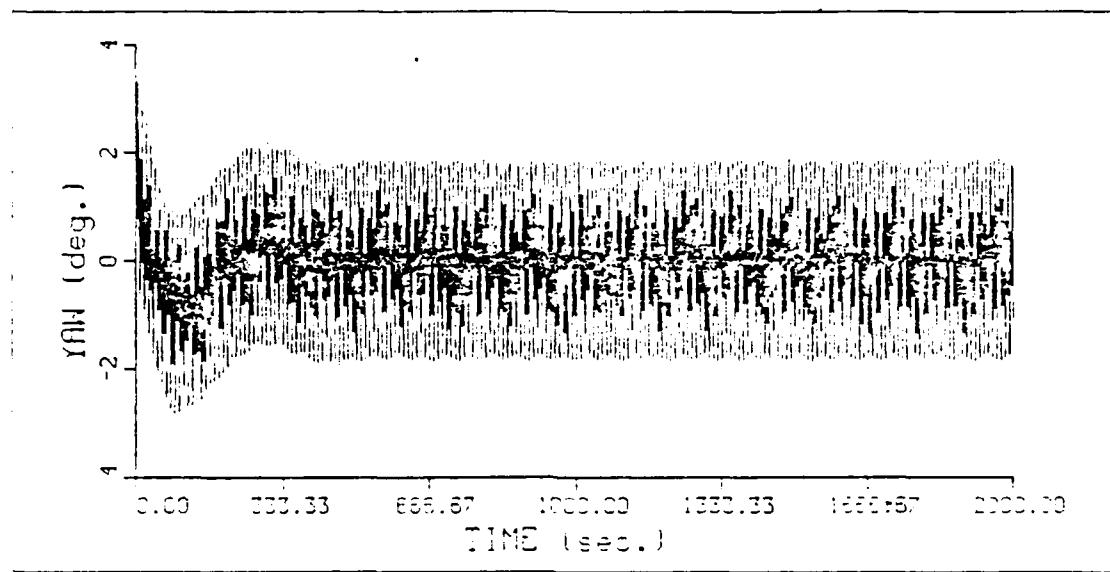
Figure 7.7   S. State=8, D Type Cont., Speed=15 Knots
Encounter Angle=120, Encounter Frequency=0.64
Yaw Degree vs. Time.
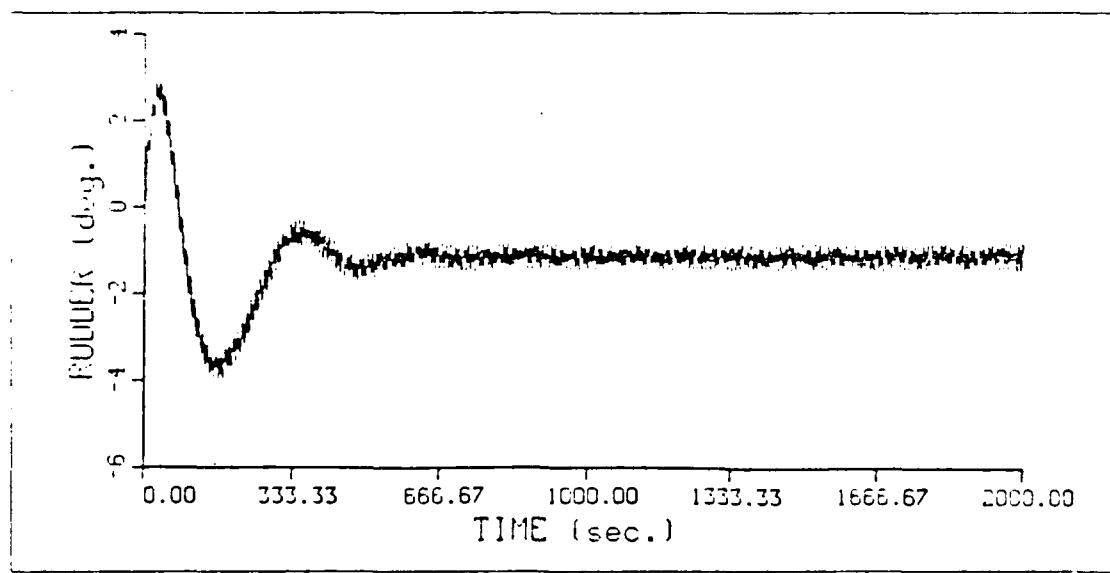
Figure 7.8   S. State=8, D Type Cont., Speed=15 Knots
Encounter Angle=120, Encounter Frequency=0.64
Rudder vs. Time.

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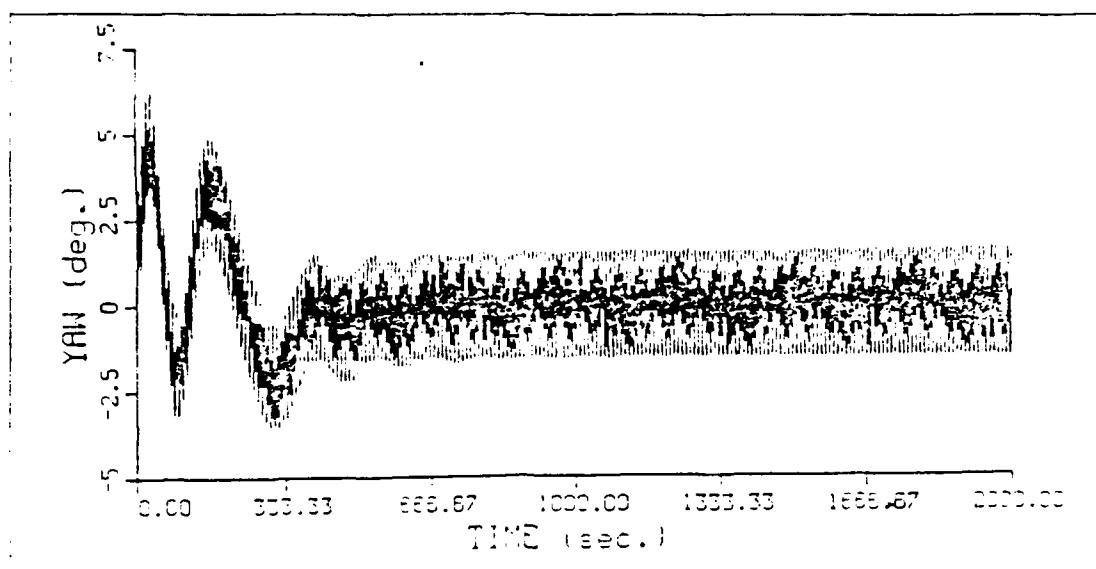
Figure 7.9   S. State=8, D Type Cont., Speed=15 Knots
Encounter Angle=150, Encounter Frequency=0.67
Yaw Degree vs. Time.
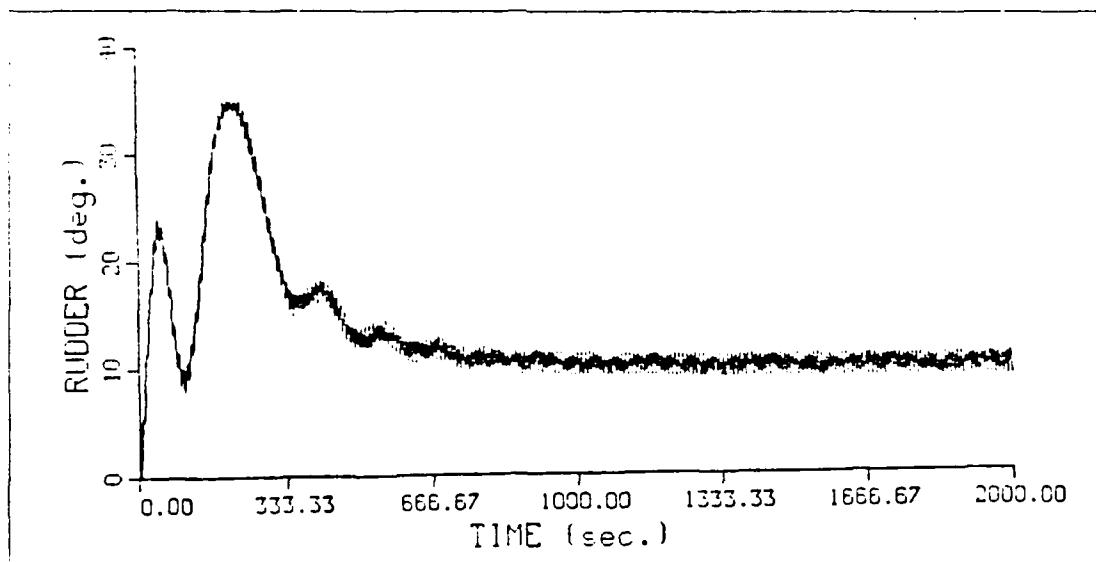
Figure 7.10   S. State=8, D Type Cont., Speed=15 Knots
Encounter Angle=150, Encounter Frequency=0.67
Rudder vs. Time.
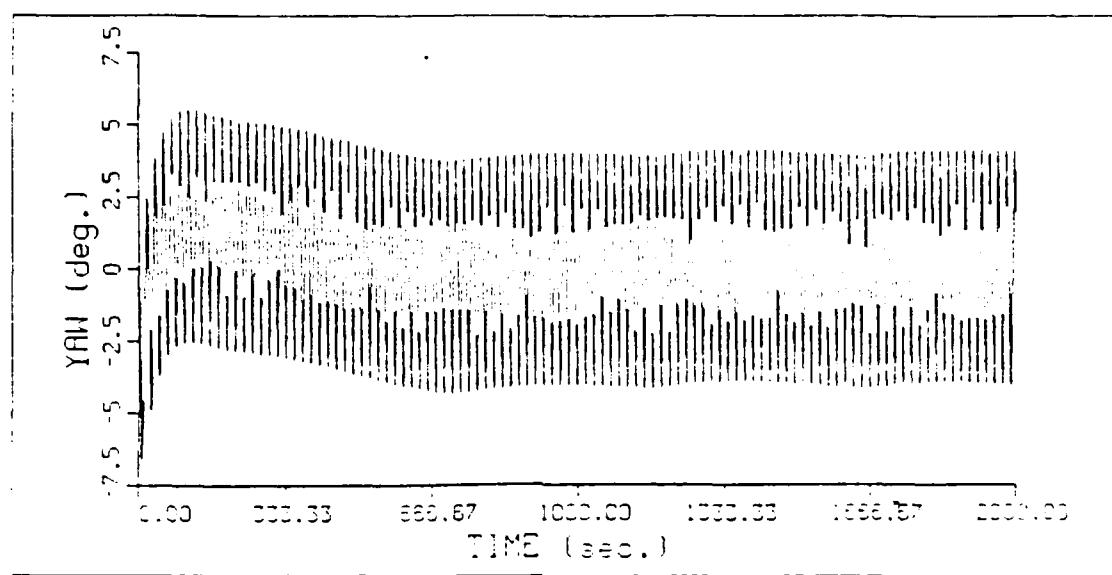
Figure 7.11   S. State=8, Optimal PID, Speed=15 Knots
Encounter Angle=030, Encounter Frequency=0.33
Yaw Degree vs. Time.
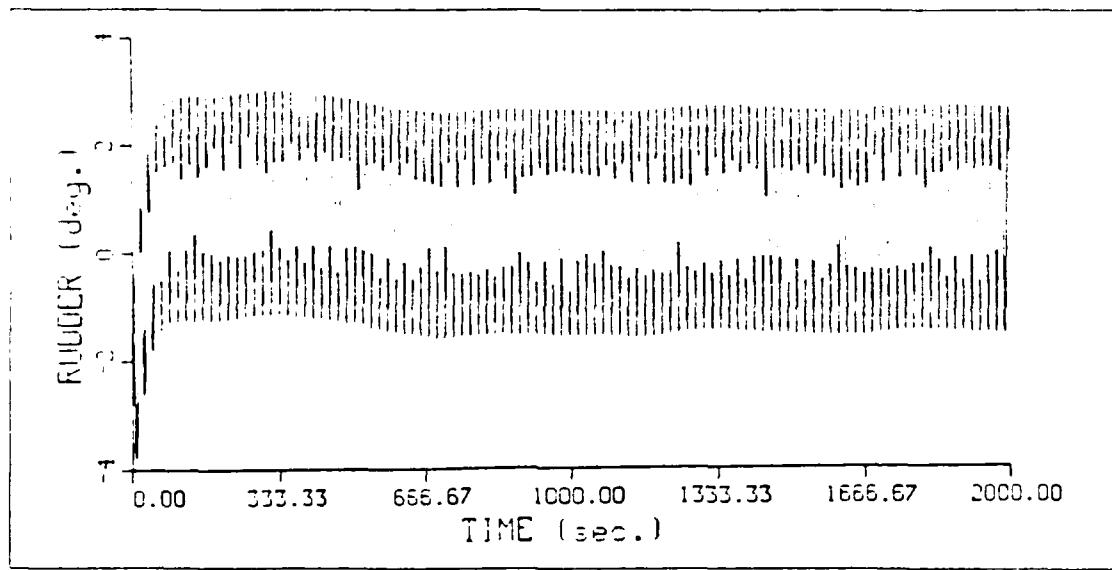
Figure 7.12   S. State=8, Optimal PID, Speed=15 Knots
Encounter Angle=030, Encounter Frequency=0.33
Rudder vs. Time.
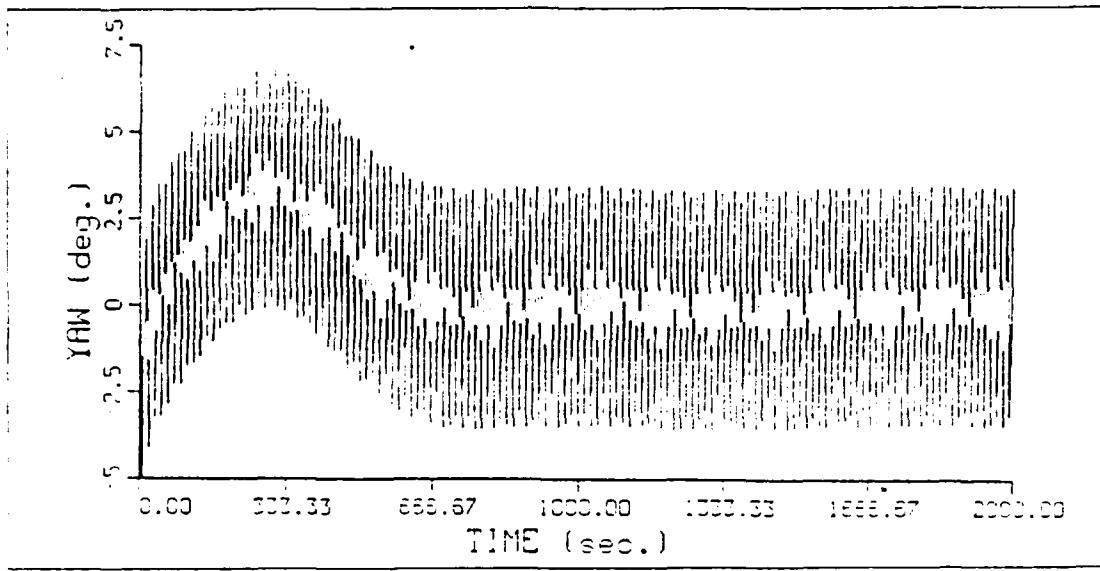
Figure 7.13   S. State=8, Optimal PID, Speed=15 Knots
Encounter Angle=060, Encounter Frequency=0.43
Yaw Degree vs. Time.
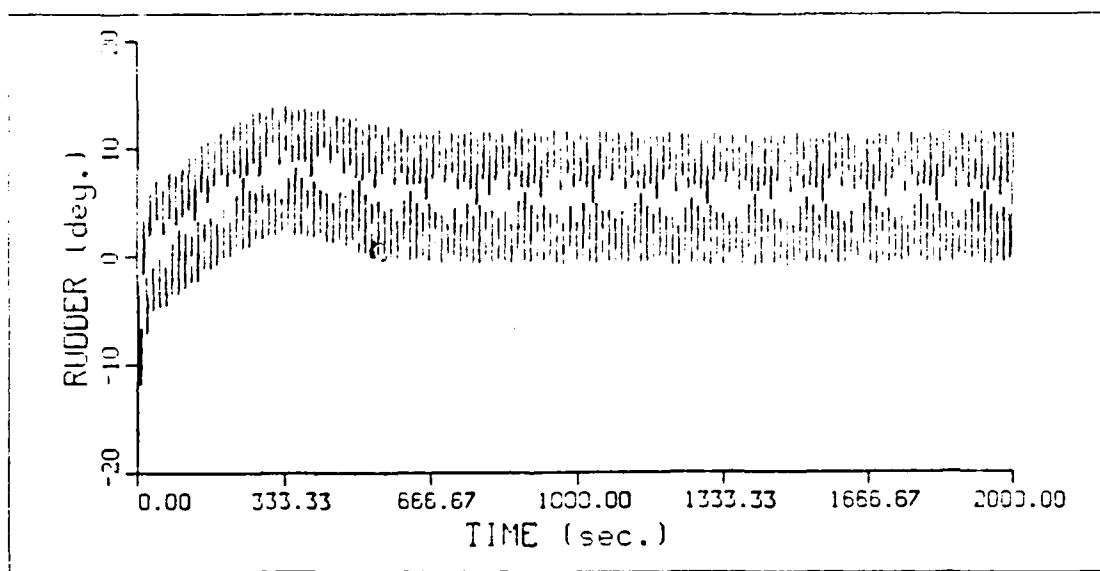
Figure 7.14   S. State=8, Optimal PID, Speed=15 Knots
Encounter Angle=060, Encounter Frequency=0.43
Rudder vs. Time.
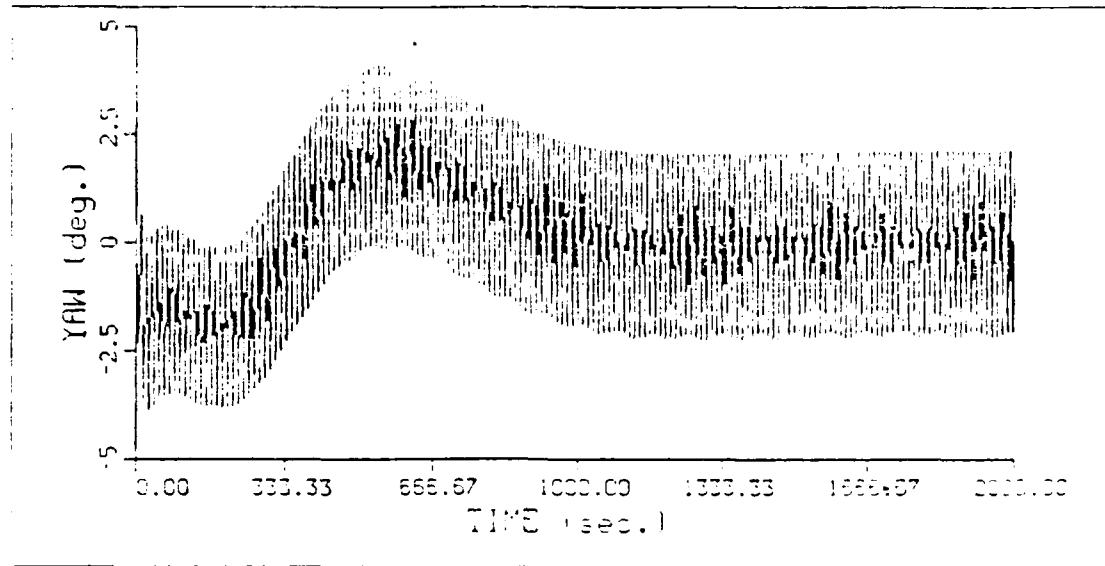
Figure 7.15   S. State=8, Optimal PID, Speed=15 Knots
Encounter Angle=090, Encounter Frequency=0.53
Yaw Degree vs. Time.
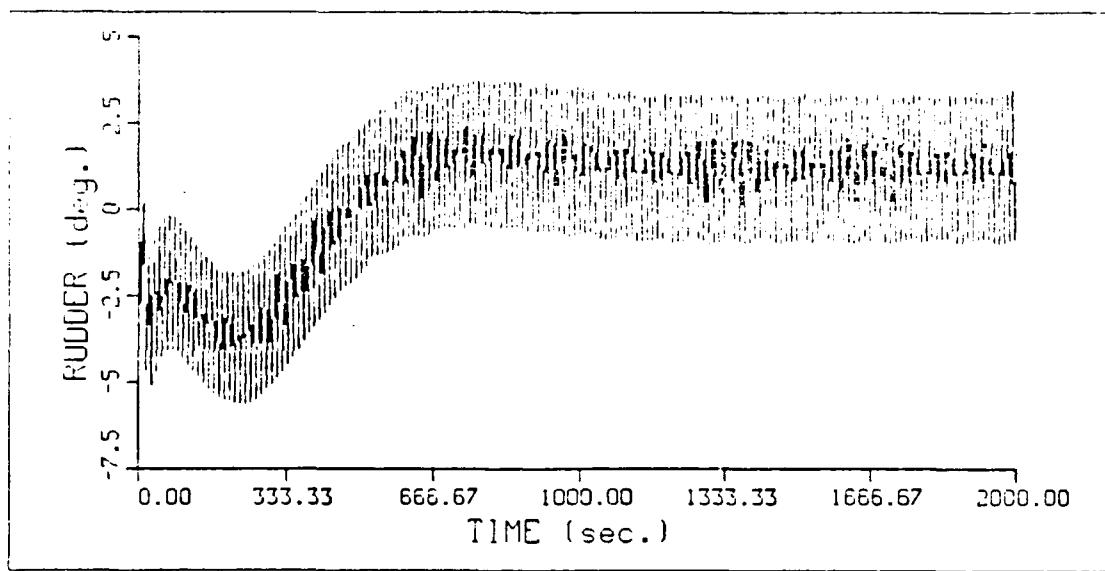
Figure 7.16   S. State=8, Optimal PID, Speed=15 Knots
Encounter Angle=090, Encounter Frequency=0.53
Rudder vs. Time.
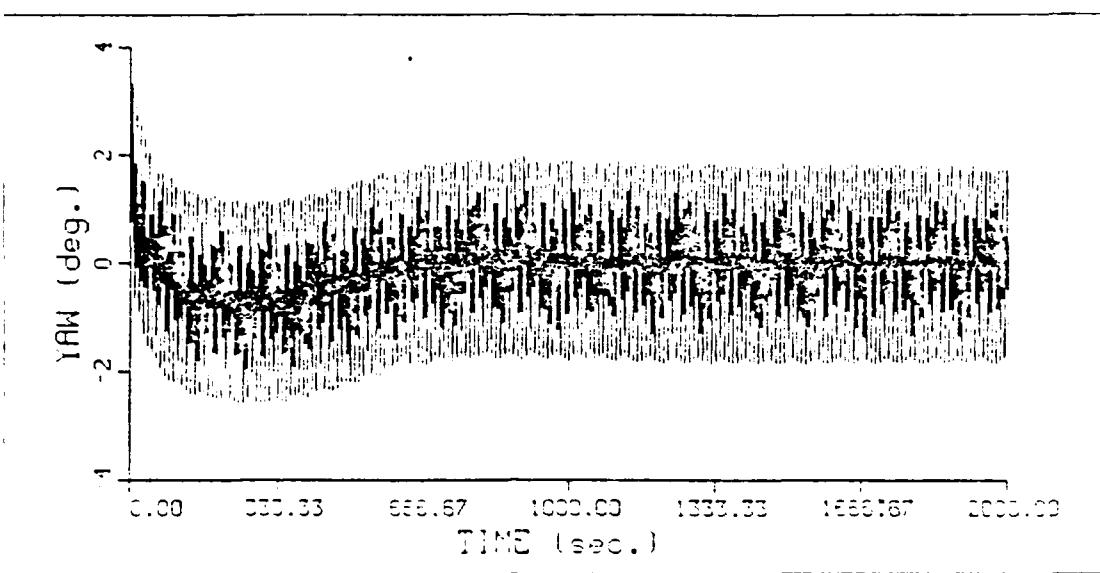
Figure 7.17   S. State=8, Optimal PID, Speed=15 Knots
Encounter Angle=120, Encounter Frequency=0.64
Yaw Degree vs. Time.
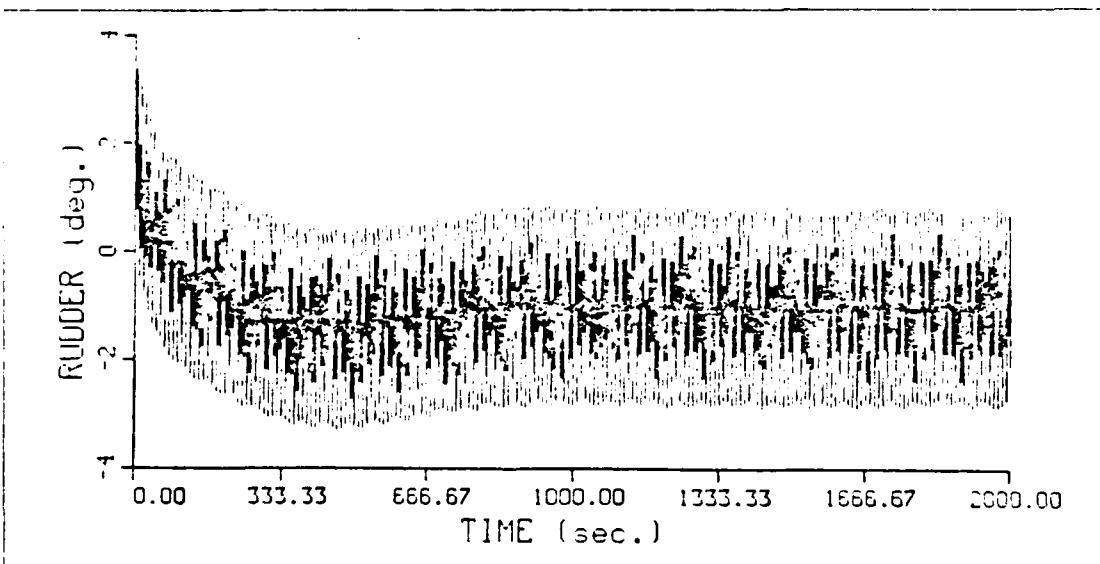
Figure 7.18   S. State=8, Optimal PID, Speed=15 Knots
Encounter Angle=120, Encounter Frequency=0.64
Rudder vs. Time.

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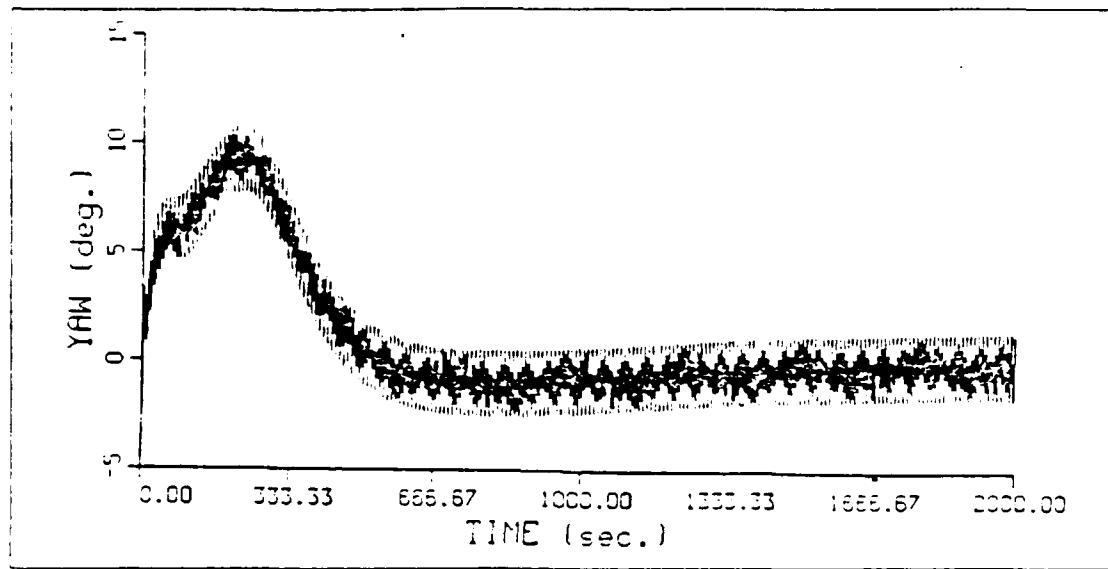
Figure 7.19   S. State=8, Optimal PID, Speed=15 Knots
Encounter Angle=150, Encounter Frequency=0.67
Yaw Degree vs. Time.
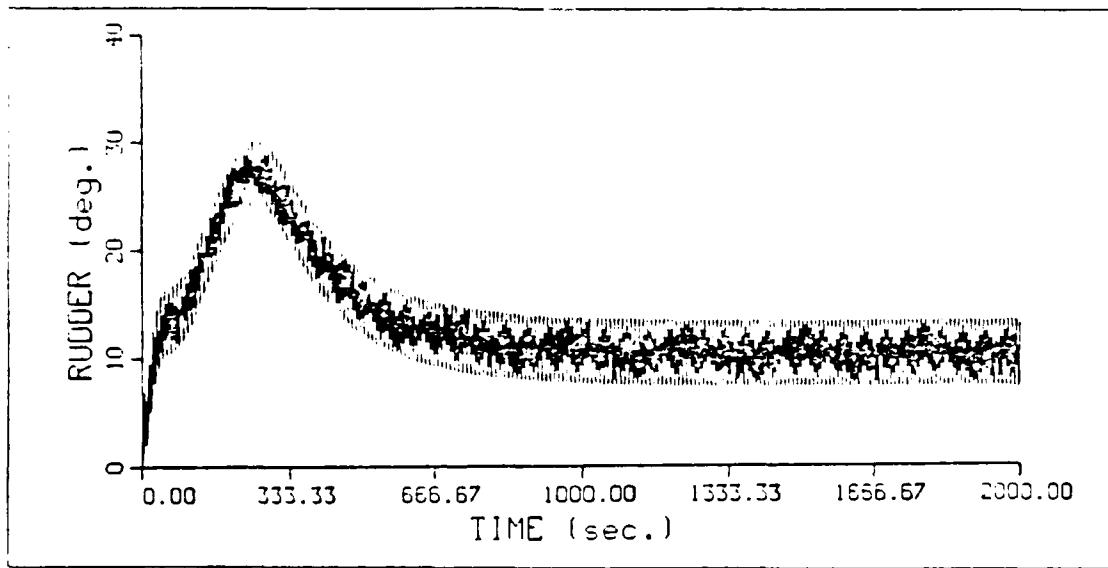
Figure 7.20   S. State=8, Optimal PID, Speed=15 Knots
Encounter Angle=150, Encounter Frequency=0.67
Rudder vs. Time.
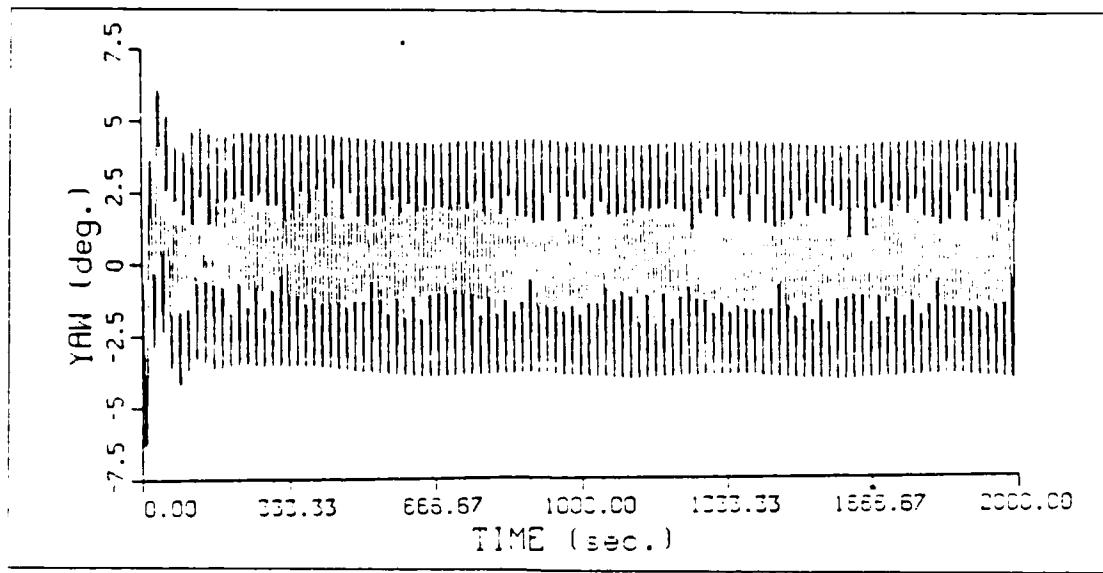
Figure 7.21   S. State=8, Well Tuned PID, Speed=15 Knots
Encounter Angle=030, Encounter Frequency=0.33
Yaw Degree vs. Time.
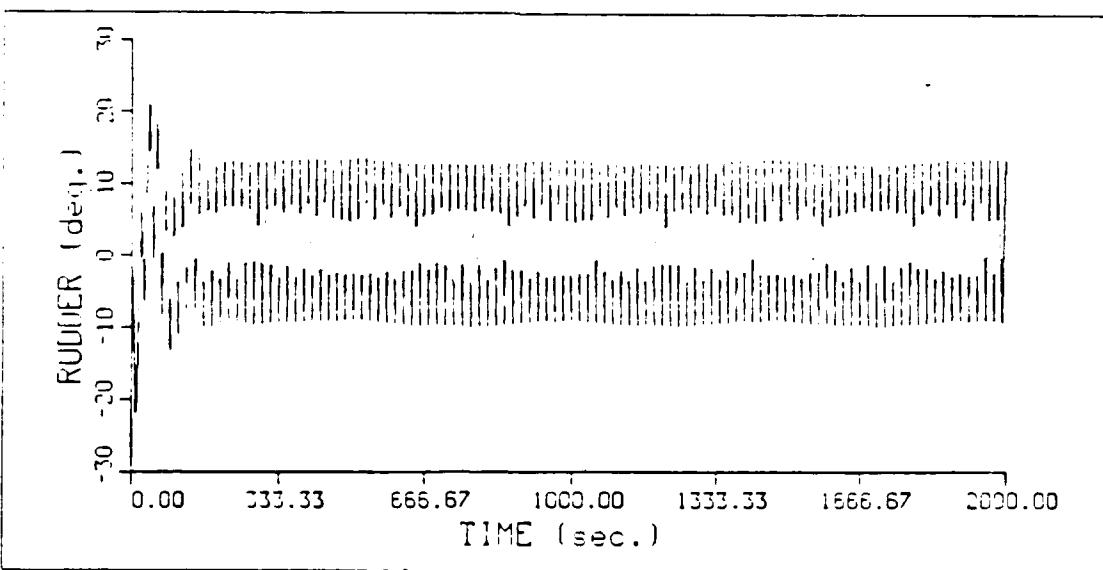
Figure 7.22   S. State=8, Well Tuned PID, Speed=15 Knots
Encounter Angle=030, Encounter Frequency=0.33
Rudder vs. Time.

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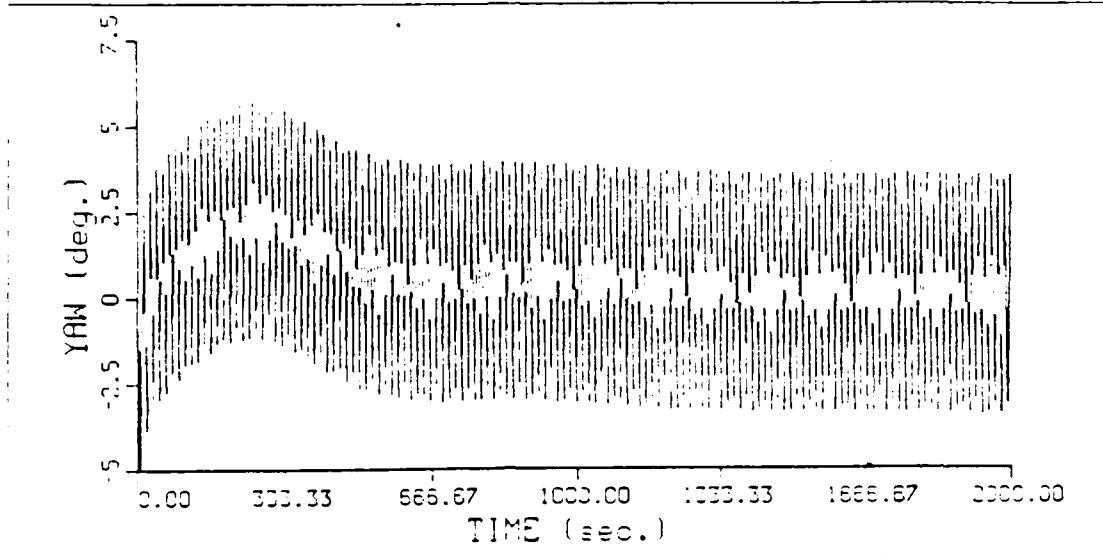
Figure 7.23   S. State=8, Well Tuned PID, Speed=15 Knots
Encounter Angle=060, Encounter Frequency=0.43
Yaw Degree vs. Time.
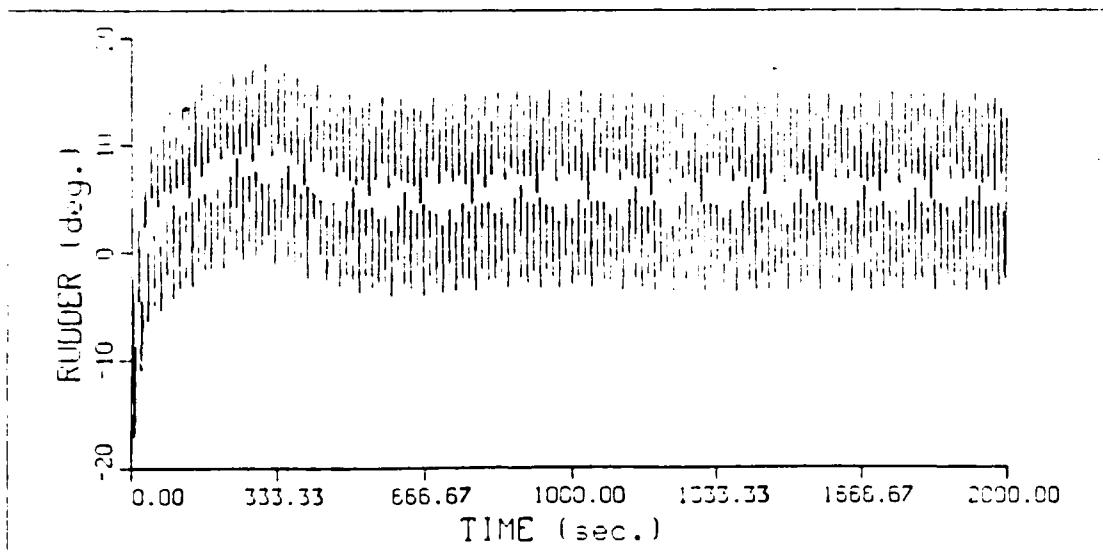
Figure 7.24   S. State=8, Well Tuned PID, Speed=15 Knots
Encounter Angle=060, Encounter Frequency=0.43
Rudder vs. Time.
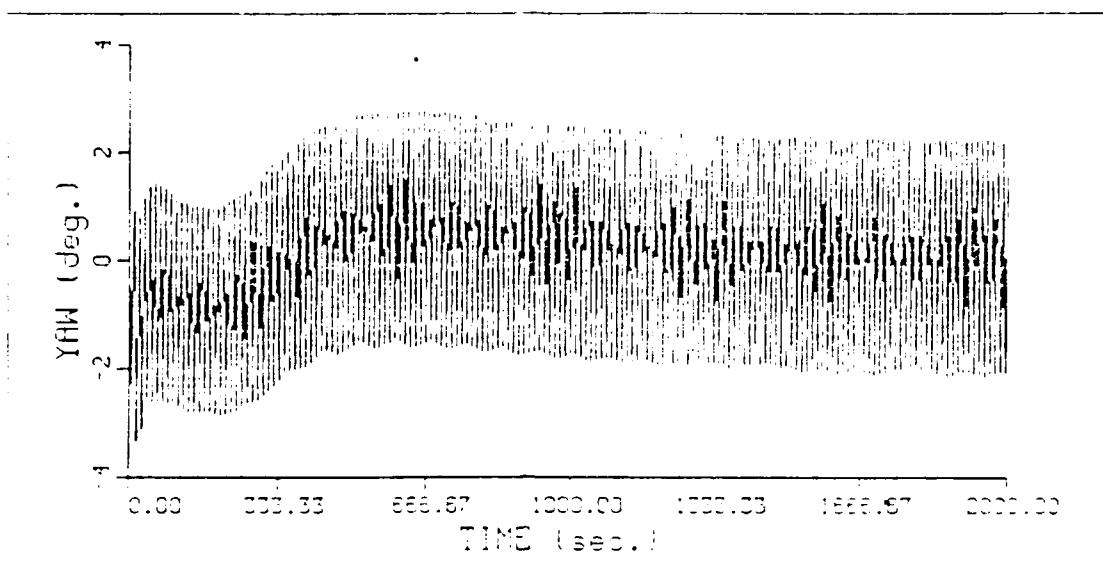
Figure 7.25. S. State=8, Well Tuned PID, Speed=15 Knots
Encounter Angle=090, Encounter Frequency=0.53
Yaw Degree vs. Time.
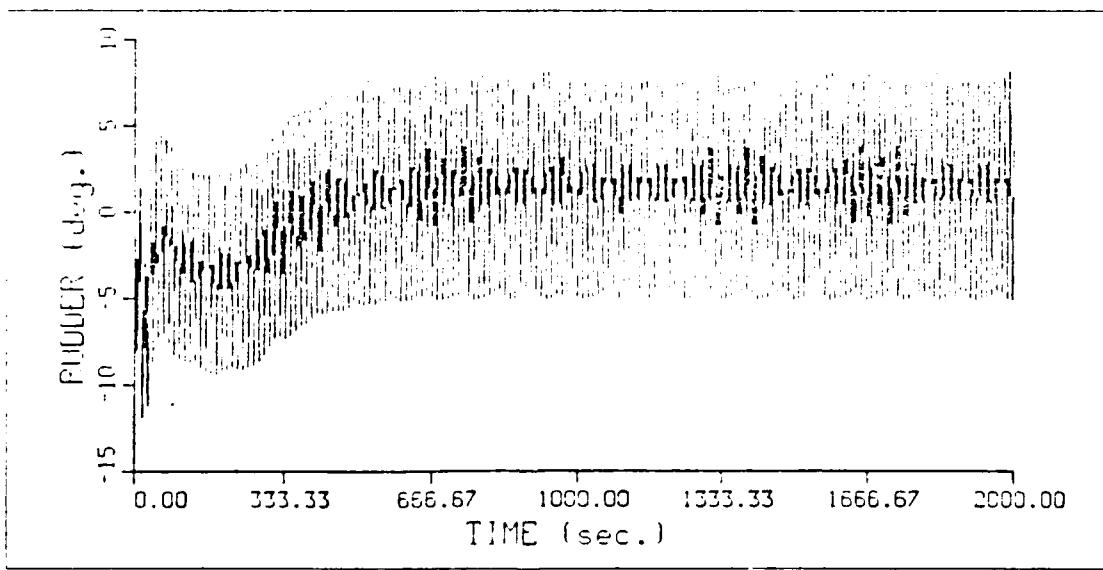
Figure 7.26   S. State=8, Well Tuned PID, Speed=15 Knots
Encounter Angle=090, Encounter Frequency=0.53
Rudder vs. Time.
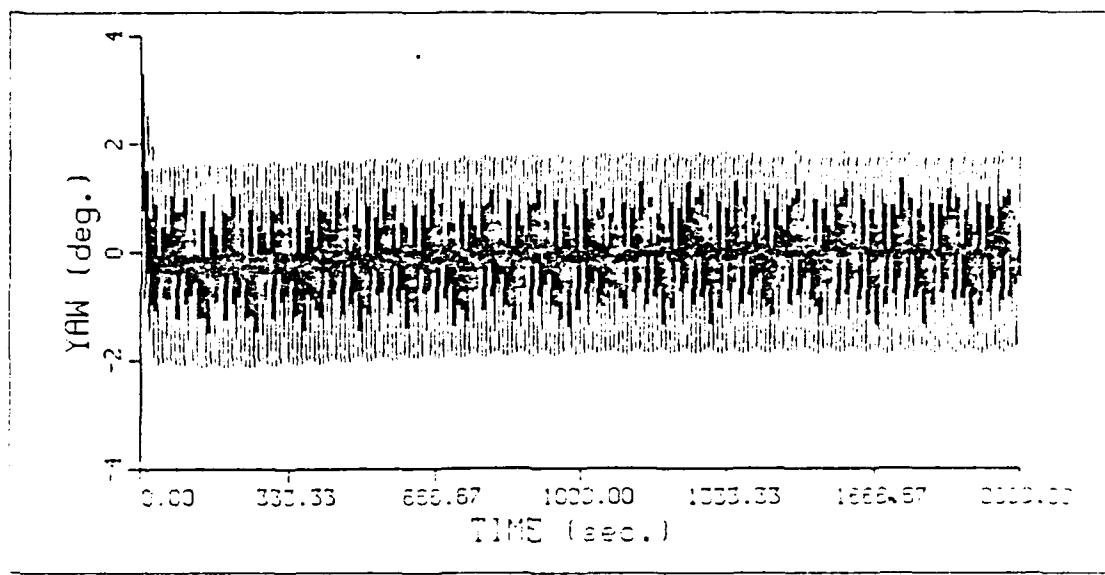
Figure 7.27   S. State=8, Well Tuned PID, Speed=15 Knots
Encounter Angle=120, Encounter Frequency=0.64
Yaw Degree vs. Time.
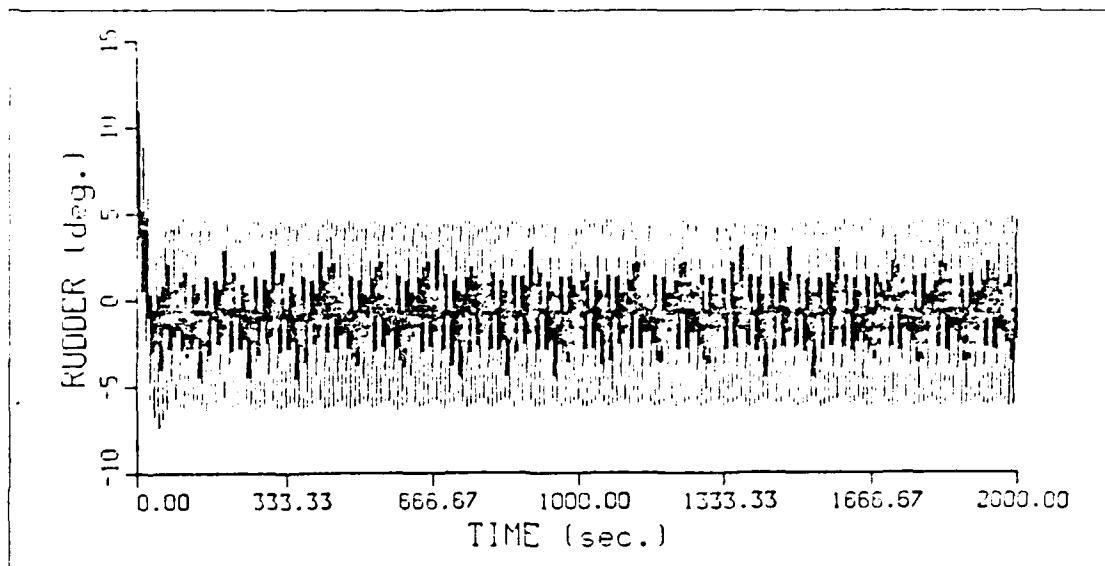
Figure 7.28   S. State=8, Well Tuned PID, Speed=15 Knots
Encounter Angle=120, Encounter Frequency=0.64
Rudder vs. Time.

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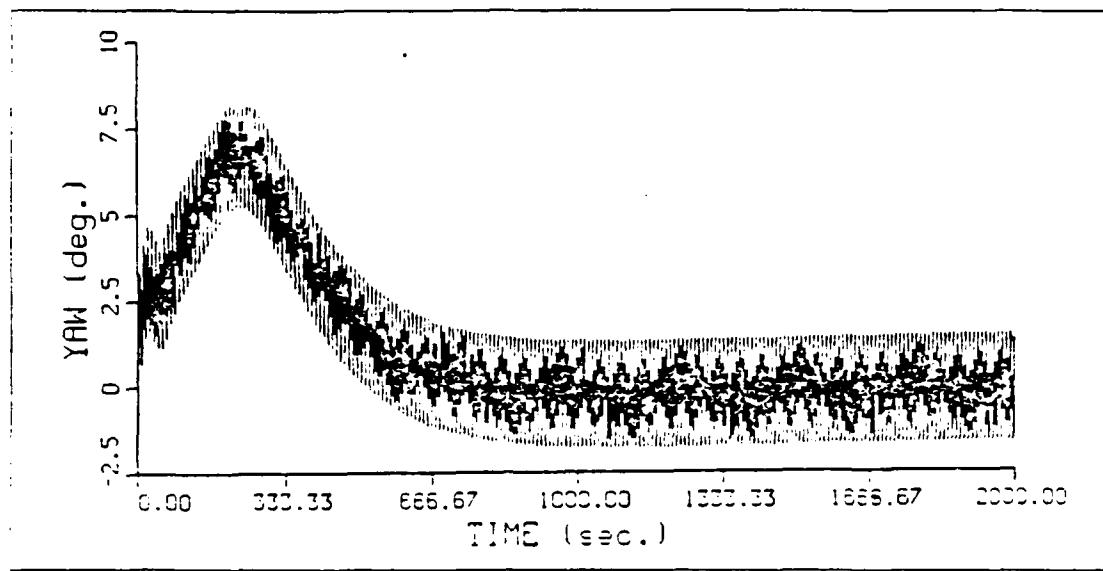
Figure 7.29   S. State=8, Well Tuned PID, Speed=15 Knots
Encounter Angle=150, Encounter Frequency=0.67
Yaw Degree vs. Time.
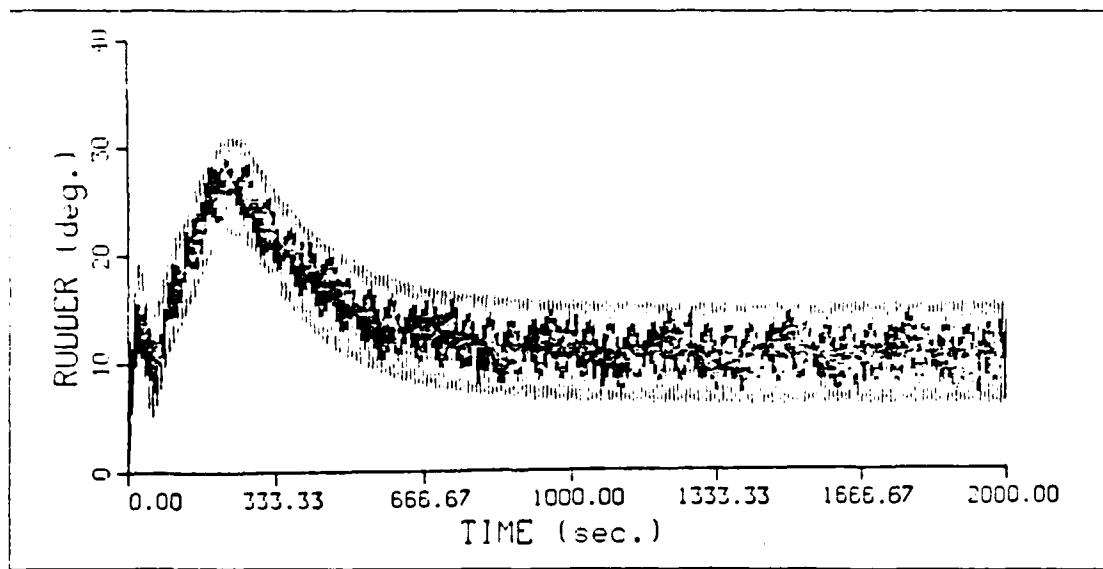
Figure 7.30   S. State=8, Well Tuned PID, Speed=15 Knots
Encounter Angle=150, Encounter Frequency=0.67
Rudder vs. Time.

VIII. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

An A Type controller was used in finding optimal parameters to minimize the propulsion losses due to steering in previous theses [Refs. 8, 10, 21, 22]. But to prevent the drifting effect of second-order forces and moments on the ship's motions, a D Type controller was used in designing the Automatic Optimal Controller. The only difference between them is the integrator part omitted in the A Type.

Savings in fuel for an Automatic Optimal Controller, an Optimal PID, and a Well-Tuned PID were compared by using a new approach explained in Chapter 6 that does not use engine specifications. It can be used in finding any ship's fuel savings by finding the added resistance due to steering.

The optimal internal and external settings of a PID to minimize the propulsion losses due to steering are found by the modified simulation program shown in Appendix C, since there is not enough information about the settings of control parameters of PID for the Mariner Class ship. The same performance criterion explained in Chapter 5 was used in finding the optimal parameters for both of them. As a result, it is found that the optimal PID is as good as Automatic Optimal Controller. They provide almost the same fuel savings in the same situations.

The well tuned PID was defined as a PID controller to reduce the heading error as fast as possible and to get fast response of the rudder to do this. Only external control settings (explained in Appendix A) are available to the operator. The optimal internal control settings were used in simulations.The Automatic Optimal Controller provides fuel savings in excess of 0.5 % over a well tuned PID when operating at sea state, 8, speed, 15 Knots, and wave frequency, 0.53 radian/seconds.

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B. RECOMMENDATIONS FOR FUTURE STUDIES

Additional work should be done to compare the Adaptive Automatic Optimal Controller with a well tuned PID for different sea states, speeds, and encounter frequencies.

In this thesis the well tuned PID was used as explained in Chapter 7. If more information is available for the PID, especially if a precise definition can be found for the well tuned PID, then the comparison in fuel savings with these well tuned PID settings will provide better insight into the value of the Optimal Controller.

In this thesis, surge, sway, and yaw equations of motion were used in simulations. The roll equation of motion may be added to the simulation program and the optimal parameters of the Automatic Optimal Controller may be found by using the BOXPLX minimization subroutine. Then the comparison in fuel savings will be more realistic.

APPENDIX A
PID CONTROLLER

The standard form of a PID controller is based on proportional-plus-integral-plus-derivative control. No provision for automatic adaptivity of the controller to speed or seaway effects exists in the PID. It relies on adjustment of the control parameters by the operator.

1. GENERAL DESCRIPTION

An autopilot commonly found aboard merchant ships has the characteristics of a PID controller. The PID control law is described by [Refs. 23, 24]:

$$\delta_c = K_1 \left[K_2 \left(1 + \frac{T_1 s}{(1+T_2 s)^2} \right) + \frac{1}{T_3 s} \right] \psi_e$$

where the gains are determined by operator interaction in the following manner:

1. Weather Adjust Gain

$K_1 = 1/3$ within "weather adjust" zone

$= 1$ outside "weather adjust" zone

The weather adjust zone is variable from 0.3 to 5.0 degree heading error. Panel is marked "0" to "5".

2. Rudder Multiplier Gain

$K_2 = 1$ to 3, continuously adjustable (panel control)

3. Internal Control Settings

The time constants T_1, T_2, and T_3 are preset for the specific ship, with the exception that may be reduced by a factor of 2 by operator interaction. The ranges of possible gyropilot time constants are:

1. Derivative Time Constant

T_1 = 2.5, 5, 10, 25, 50, 100, 150, 200 second

"Rate Mult" control reduces the time constant by a factor of 2 when at minimum setting.

2. Derivative Filter Time Constant

T_2 = 1.5, 2.5, 5, 10, 15 second

3. Integral Time Constant

T_3 = 200, 400, 600, 1000 second

2. APPLIED TO SIMULATION PROGRAM

Figure A.1 corresponds to controller PID which has the form [ref. 22]:

$$K_1 \left[K_2 \left(1 + \frac{T_1 s}{(1+T_2 s)^2} \right) + \frac{1}{T_3 s} \right]$$

This can be written in simulation program as this:

```
DX2 = ( YAWE - X2 ) / T_2
DX3 = ( X2 - X3 ) / T_2
D = ( K_1*K_2*T_1*DX3 ) + ( K_1*K_2*YAWE) + X4
```

Where

```
X2 = X2 + ( DX2 * DELT )
X3 = X3 + ( DX3 * DELT )
X4 = X4 + ( YAWE * (K_1/T_3) ) * DELT
```
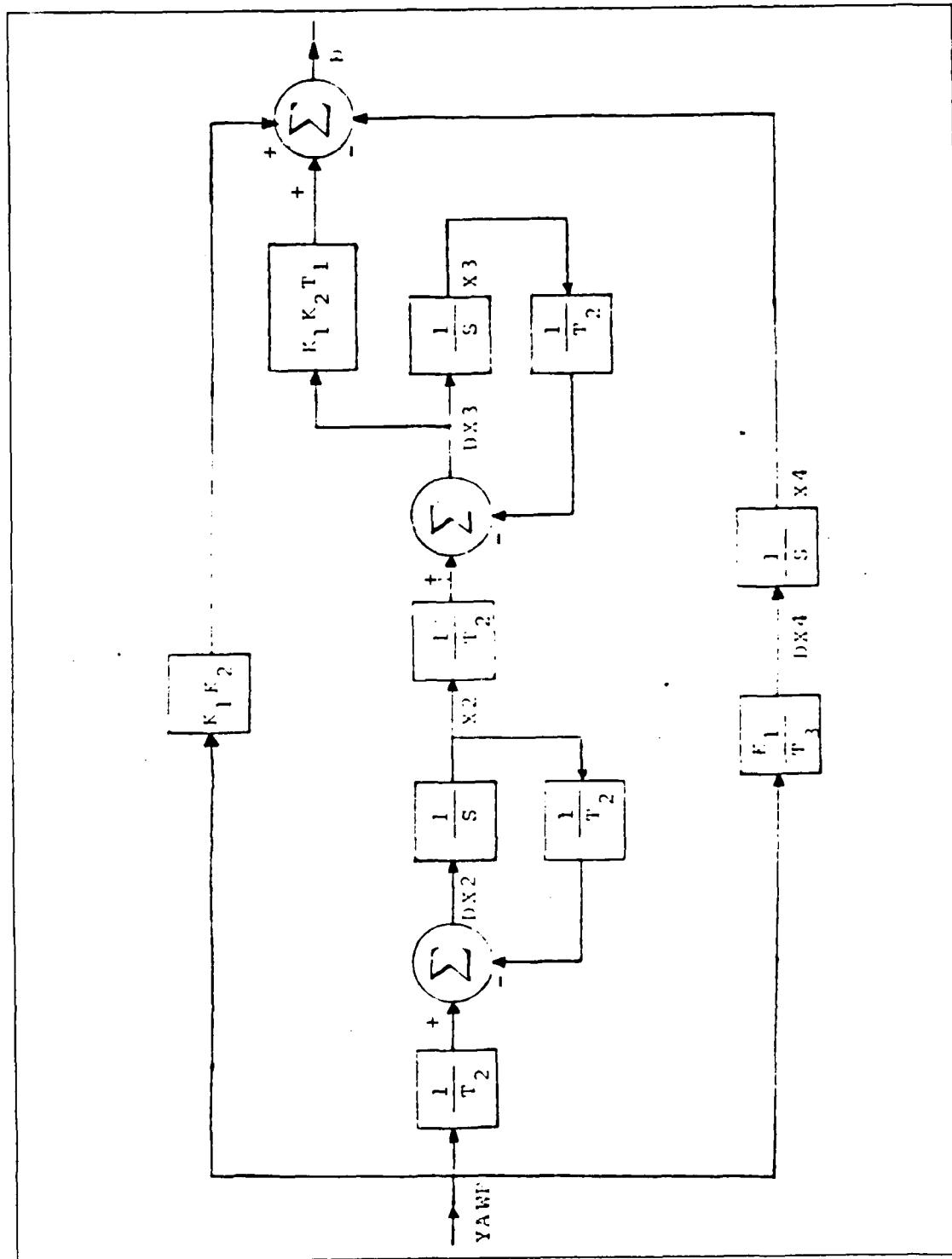
Figure A.1   PID Controller in Simulation Program.

APPENDIX B

NONLINEAR EQUATIONS OF MOTION AND COEFFICIENTS TABLES

Nonlinear Equations of Motions:

X-Eq.: $(m - X_{\dot{u}})\dot{u} = f_1(u,v,r,\delta)$

Y-Eq.: $(m - Y_{\dot{v}})\dot{v} + (mX_G - Y_{\dot{r}})\dot{r} = f_2(u,v,r,\delta)$ (eqn B.1)

N-Eq.: $(mX_G - N_{\dot{v}})\dot{v} + (I_z - N_{\dot{r}})\dot{r} = f_3(u,v,r,\delta)$

Where

$f_1(u,v,r,\delta) = X^* + X^*_u\Delta u + (1/2)X_{uu}\Delta u^2 + (1/6)X_{uuu}\Delta u^3 +$

$(1/2)X_{vv}v^2 + ((1/2)X_{rr}+mX_G)r^2 + (1/2)X_{\delta\delta}\delta^2 +$

$(1/2)X_{vvu}v^2\Delta u + (1/2)X_{rru}r^2\Delta u + (1/2)X_{\delta\delta u}\delta^2\Delta u +$

$(X_{vr}+m)vr + X_{v\delta}v\delta + X_{r\delta}r\delta + X_{vru}vr\Delta u +$

$X_{v\delta u}v\delta\Delta u + X_{r\delta u}r\delta\Delta u$

$$f_2(u,v,r,\delta) = Y* + Y*_u \Delta u + Y*_{uu} \Delta u^2 + Y_v v + (1/6)Y_{vvv}v^3 +$$

$$(1/2)Y_{vrr}vr^2 + (1/2)Y_{v\delta\delta}v\delta^2 + Y_{vu}v\Delta u +$$

$$(1/2)Y_{vuu}v\Delta u^2 + (Y_r - mu)r + (1/6)Y_{rrr}r^3 +$$

$$(1/2)Y_{rvv}rv^2 + (1/2)Y_{r\delta\delta}r\delta^2 + Y_{ru}r\Delta u +$$

$$(1/2)Y_{ruu}r\Delta u^2 + Y_\delta \delta + (1/6)Y_{\delta\delta\delta}\delta^3 + .$$

$$(1/2)Y_{\delta vv}\delta v^2 + Y_{\delta u}\delta\Delta u + (1/2)Y_{\delta uu}\delta\Delta u^2 +$$

$$Y_{vr\delta}vr\delta + (1/2)Y_{\delta rr}\delta r^2$$

$$f_3(u,v,r,\delta) = N* + N*_u \Delta u + N*_{uu} \Delta u^2 + N_v v + (1/6)N_{vvv}v^3 +$$

$$(1/2)N_{vrr}vr^2 + (1/2)N_{v\delta\delta}v\delta^2 + N_{vu}v\Delta u +$$

$$(1/2)N_{vuu}v\Delta u^2 + (N_r - mX_G u)r + (1/6)N_{rrr}r^3 +$$

$$(1/2)N_{rvv}rv^2 + (1/2)N_{r\delta\delta}r\delta^2 + N_{ru}r\Delta u +$$

$$(1/2)N_{ruu}r\Delta u^2 + N_\delta \delta + (1/6)N_{\delta\delta\delta}\delta^3 +$$

$$(1/2)N_{\delta vv}\delta v^2 + N_{\delta u}\delta\Delta u + (1/2)N_{\delta uu}\delta\Delta u^2 +$$

$$N_{vr\delta}vr\delta + (1/2)N_{\delta rr}\delta r^2$$

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TABLE VI

COEFFICIENTS OF X-EQUATION

Taylor Expansion and Dynamic response Terms	Identifier in Fortran Program	Nondim. Factor	Nondim. Coeff. $*10^5$
$(m-X_{\dot{u}})$	C1	$(1/2)\rho\ LBP^3$	840.0
X_u	X1	$(1/2)\rho\ LBP^2 S\cdot$	-120.0
$(1/2)X_{uu}$	X2	$(1/2)\rho\ LBP^2$	45.0
$(1/6)X_{uuu}$	X3	$(1/2)\rho\ LBP^2/S$	-10.3
$(1/2)X_{vv}$	X4	$(1/2)\rho\ LBP^2$	-898.8
$(1/2)X_{rr}+mX_G$	X5	$(1/2)\rho\ LBP^4$	18.0
$(1/6)X_{\delta\delta}$	X6	$(1/2)\rho\ LBP^2 S^2$	-94.8
$(1/2)X_{vvu}$		$(1/2)\rho\ LBP^2/S$	
$(1/2)X_{rru}$		$(1/2)\rho\ LBP^4/S$	
$(1/2)X_{\delta\delta u}$		$(1/2)\rho\ LBP^2 S$	
$(X_{vr}+m)$	X7	$(1/2)\rho\ LBP^3$	798.0
$X_{v\delta}$	X8	$(1/2)\rho\ LBP^2 S$	83.2
$X_{r\delta}$		$(1/2)\rho\ LBP^3 S$	0.0
X_{vru}		$(1/2)\rho\ LBP^3/S$	
$X_{v\delta u}$		$(1/2)\rho\ LBP^2$	
$X_{r\delta u}$		$(1/2)\rho\ LBP^3$	
$X*$	X0	$(1/2)\rho\ LBP^2 S^2$	0.0

TABLE VII

COEFFICIENTS OF Y-EQUATION

Taylor Expansion and Dynamic response Terms	Identifier in Fortran Program	Nondim. Factor	Nondim. Coeff. $*10^5$
$(m-Y_{\dot{v}})$	C2	$(1/2)\rho\ LBP^3$	1546.0
$(mX_G-Y_{\dot{r}})$	C3	$(1/2)\rho\ LBP^4$	-8.6
Y_v	Y1	$(1/2)\rho\ LBP^2S$	-1160.4
$(1/6)Y_{vvv}$	Y2	$(1/2)\rho\ LBP^2/S$	-8078.2
$(1/2)Y_{vrr}$		$(1/2)\rho\ LBP^4/S$	0.0
$(1/2)Y_{v\delta\delta}$	Y3	$(1/2)\rho\ LBP^2S$	-3.8
(Y_r-mu)	Y4	$(1/2)\rho\ LBP^3S$	-499.0
$(1/6)Y_{rrr}$		$(1/2)\rho\ LBP^5/S$	0.0
$(1/2)Y_{rvv}$	Y5	$(1/2)\rho\ LBP^3/S$	15356.0
$(1/2)Y_{r\delta\delta}$		$(1/2)\rho\ LBP^3S$	0.0
Y_δ	Y6	$(1/2)\rho\ LBP^2S^2$	277.9
$(1/6)Y_{\delta\delta\delta}$	Y7	$(1/2)\rho\ LBP^2S^2$	-90.0
$(1/2)Y_{\delta vv}$	Y8	$(1/2)\rho\ LBP^2$	1189.6
$(1/2)Y_{\delta rr}$		$(1/2)\rho\ LBP^4$	0.0
$Y_{\delta u}$		$(1/2)\rho\ LBP^2S$	0.0
$(1/2)Y_{\delta uu}$		$(1/2)\rho\ LBP^2$	
$Y_{vr\delta}$		$(1/2)\rho\ LBP^3$	0.0
Y^*	Y0	$(1/2)\rho\ LBP^2S^2$	-3.6
Y^*_u		$(1/2)\rho\ LBP^2S$	0.0

TABLE VIII
COEFFICIENTS OF N-EQUATION

Taylor Expansion and Dynamic response Terms	Identifier in Fortran Program	Nondim. Factor	Nondim. Coeff. $*10^5$
$(mX_G - N_{\dot{v}})$	C5	$(1/2)\rho\ LBP^4$	-22.7
$(I_z - N_{\dot{r}})$	C4	$(1/2)\rho\ LBP^5$	82.9
N_v	N1	$(1/2)\rho\ LBP^3 S$	-263.5
$(1/6)N_{vvv}$	N2	$(1/2)\rho\ LBP^3/S$	1636.1
$(1/2)N_{vrr}$		$(1/2)\rho\ LBP^5/S$	0.0
$(1/2)N_{v\delta\delta}$	N3	$(1/2)\rho\ LBP^3 S$	12.5
$(N_r - mX_G u)$	N4	$(1/2)\rho\ LBP^4 S$	-166.0
$(1/6)N_{rrr}$		$(1/2)\rho\ LBP^6/S$	0.0
$(1/2)N_{rvv}$	N5	$(1/2)\rho\ LBP^4/S$	-5483.0
$(1/2)N_{r\delta\delta}$		$(1/2)\rho\ LBP^4 S$	0.0
N_δ	N6	$(1/2)\rho\ LBP^3 S^2$	-138.8
$(1/6)N_{\delta\delta\delta}$	N7	$(1/2)\rho\ LBP^3 S^2$	45.0
$(1/2)N_{\delta vv}$	N8	$(1/2)\rho\ LBP^3$	-489.0
$(1/2)N_{\delta rr}$		$(1/2)\rho\ LBP^4 S$	0.0
$N_{\delta u}$		$(1/2)\rho\ LBP^3 S$	0.0
$(1/2)N_{vr\delta}$		$(1/2)\rho\ LBP^4$	0.0
N^*	N0	$(1/2)\rho\ LBP^3 S^2$	2.8
N^*_u		$(1/2)\rho\ LBP^3 S$	0.0

SIMULATION PROGRAM

```
C***   DECLERATIONS OF VARIABLES   **************************
       REAL   MH(10000),T(10000)
       REAL*8 L,L2,L3,L4,L5,L6
       REAL*8 X,XDOT,Y,YDOT,U,UDOT,V,VDOT,YAW,R,RDOT
       REAL*8 X1,X2,X3,X4,X5,X6,X7,X8
       REAL*8 Y0,Y1,Y2,Y3,Y4,Y5,Y6,Y7,Y8
       REAL*8 N0,N1,N2,N3,N4,N5,N6,N7,N8,TEM,TEMP,TEM1
       C        ,TEMP1
       REAL*8 C1,C2,C3,C4,C5,F1,F2,F3,F2X,F2Y,M2Z,FXX,FYY
       C        ,MZZ
       REAL*8 RO,DELT,S,DU,U1,K,Z1,Z2,P1,P2,KD,RDIF
       REAL*8 DYAWE,YAWE,YAWC,ISR,ISE,TDIFF,LAMDA,RPM,RES
       C        ,EHP
       REAL*8 S1,S2,DS1,DS2,D,LOST,YAWDEG,DDEG,S3
       REAL*8 MASS,IZ,XG,YVDOT,NVDOT,YR,YRDOT,NR,NRDOT,FX
       C        ,FY,MZ
       REAL*8 RX,RY,RZ,TX,TY,TZ,WA,WE,RXR,RYR,RXI,RYI,MZR
       C        ,MZI
       REAL*8 MAGMAX,MAGMIN,TIME,ETIME,DELST1,DELST2,DELSTE
       REAL*8 BARFUE,LBFUEL
       REAL*8    TOTFR1,TOTFR2,PERC1,PERC2
C***WHEN PID CONTROLLER IS USED , ADD THE FOLLOWINGS ******
C***VARIABLES                                        ******
C      REAL*8 PIDV2,PIDV3,PIDV4,PIDDV2,PIDDV3,PIDT1,PIDT2
C    C        ,pIDT3,PIDK1,pidk2
C***WHEN OPTIMAL PID PARAMETERS ARE WANTED TO FIND,*****
C***ADD THE FOLLOWING VARIABLES AND GIVEN DATA VALUES***
C      REAL*8 KK1(2),KK2(5),TT1(9),TT2(5),TT3(4)
C      DATA KK1(1)/0.33/1./
C      DATA KK2(1),KK2(2),KK2(3),KK2(4),KK2(5)/1.,1.5,2.
```

```
C     C                                             ,2.5,3./
C       DATA TT1(1),TT1(2),TT1(3),TT1(4),TT1(5),TT1(6)
C     C       ,TT1(7),TT1(8),
C     C       TT1(9)/2.5,5.,10.,15.,25.,50.,100.,150.,200./
C       DATA TT2(1),TT2(2),TT2(3),TT2(4),TT2(5)/1.5,2.5,5.
C     C                                             ,10.,15./
C       DATA TT3(1),TT3(2),TT3(3),TT3(4)/200.,400.,600.
C     C                                   .,1000./
C*********************************************************
C
C
C*** THESE DISSPLA STATEMENTS ARE USED ONLY WHEN ******
C*** RUNNING TO GET GRAPHS.  THEY ARE NOT USED     ******
C*** WHEN FINDING OPTIMAL  PARAMETERS FOR PID.     ******
        CALL TEK618
        CALL PAGE(11.0,8.5)
        CALL NOBRDR
        CALL HEIGHT(0.15)
        CALL MX1ALF('STANDARD','&')
        CALL MX2ALF('L/CSTD','#')
C***************************************************
C
C***   USE THIS PART TO FIND OPTIMAL PID PARAMETERS   ******
C       DO 11 IK1=1,2
C       DO 22 IK2=1,5
C       DO 33 IT1=1,9
C       DO 44 IT2=1,5
C       DO 55 IT3=1,4
C
C
C*** WHEN FINDING OPTIMAL PID PARAMETERS, ******
C*** USED ETIME=1000.                      ******
        ETIME=2000.
        TIME=0.
        ITIME=0
```

```
      ICOUNT=1
C
C
C*** THESE ARE OPTIMAL PARAMETERS OF AUTOMATIC OPTIMAL  ***
C*** CONTROLLER FOUND BY RUNNING OPTIMAL GAIN PROGRAM    ***
C*** IN REFERANS 7, 9, 21, 22, 23.                       ***
      K=0.6983
      Z1=0.0102   .    .
      P1=34.3839
      KD=569.1469
C*** HERE, YOU CAN USE WELL TUNED PID PARAMETERS OR   ****
C*** OPTIMAL PID PARAMETERS FOUND BY RUNNING THIS     ****
C*** SIMULATION PROGRAM USING THE PART FOR FINDING    ****
C*** OPTIMAL PID VALUES.                              ****
C      PIDK1=1.0
C      PIDK2=3.0
C      PIDT1=25.
C      PIDT2=15.
C      PIDT3=200.
C
C*** INITIALIZE ALL THE NECESSARY VARIABLES **************
C   INITIALIZE THE COST FUNCTION
      ISE=0.0
      ISR=0.0
      TDIFF=0.0
      LAMDA=5.48
C   X,XDOT,Y,YDOT ARE FIX COORDINATES ON EARTH
      X=0.0
      Y=0.0
      XDOT=0.0
      YDOT=0.0
C   U,UDOT,V,VDOT ARE FIX COORDINATES ON SHIP
      V=0.0
      UDOT=0.0
      VDOT=0.0
```

```
          F1=0.0
          F2=0.0
          F3=0.0
          YAW=0.0
          YAWE=0.0
          R=0.0
          RDOT=0.0
          RPM=68.6
          RES=0.
          EHP=0.0
          BARFUE = 0.0
          LBFUEL = 0.0
          RO=1.9876*.5
          G=32.174
          YAWE=0.0
          TEM=0.0
          TEMP=0.0
          TEM1=0.0
          TEMP1=0.0
          DS1=0.0
          DS2=0.0
          S1=0.0
          S2=0.0
          PIDDV2=0.0
          PIDDV3=0.0
          PIDV2=0.0
          PIDV3=0.0
          PIDV4=0.0
          SMALL=99999999.9
C    D = RUDDER ANGLE
          D=0.0/57.296
C    ORDERED SPEED IN  FEET/SEC
C     15.*1.689 FT/SEC=15 KNOTS
          U1=15.*1.689
C    AT STEADY STATE ACTUAL SPEED (U) = COMMAND SPEED (UC)
```

```
          U=U1
          L=528.
          L2=L**2
          L3=L*L*L
          L4=L*L3
          L5=L*L4
          L6=L*L5
C      FORCES IN X,Y DIRECTION COMPUTED IN FORCES
C      MOMENTS IN Z
          FX=0.
          FY=0.
          MZ=0.
C
C
C*** SEA STATE FORCES AND MOMENTS FOUND BY RUNNING*****
 THE JIM CASS'S SEA STATE PROGRAM IN REFERANS 8.   *****
C***   FIRST ORDER FORCES AND MOMENT
          RXR=-.12887D4
          RXI=-.90512D5
          RYR= .74476D3
          RYI= .11220D6
          MZR=-.20222D8
          MZI=-.10028D7
C***   SECOND ORDER FORCES  AND  MOMENT
          F2X= .90755D-3
          F2Y= .16626D-4
          M2Z= .96895D-4
C
C*** FINDING FORCES AND MOMENT MAGNITUDES AND PHASES   *****
          RX=(RXR**2+RXI**2)**.5
          RY=(RYR**2+RYI**2)**.5
          RZ=(MZR**2+MZI**2)**.5
          TX=DATAN2(RXI,RXR)
          TY=DATAN2(RYI,RYR)
          TZ=DATAN2(MZI,MZR)
```

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```
          FXX=2.*RO*G*L*WA**2*F2X
          FYY=2.*RO*G*L*WA**2*F2Y
          MZZ=2.*RO*G*(L**2)*(WA**2)*M2Z
C
C
C*** SEA STATE AND SHIP SPECIFICATIONS  ******************
C       SIGNIFICANT WAVE HEIGHT:SEA STATE 8-19 FEET, 7-15
C                           .   .          6-11, 5-7
          WA=19.
C       ENCOUNTER FREQUENCY:(WHEN ENCOUNTER ANGLE IS 30)
          WE=0.33
C       ADDED MASS AND ADDED INERTIA TERMS:
          MASS=.11155D+07
          IZ=0.000392*RO*L5
          XG=-11.96
C
C   .
   200 CONTINUE
C  INPUT YAW COMMAND
          YAWC= 0.0/57.296
C
C  TO PREVENT THE SHIP TURNING MORE THAN ONE DEGREE
C  PER SECOND AT THE BEGINNING OF THE PROGRAM SINCE
C  THE SHIP CANNOT BE IN HIGHER SEA STATE INSTANTLY.
C  THERE MUST BE TIME PERIOD C  FROM GOING SEA STATE
C  ONE TO SEA STATE EIGHT.
          TEM1=YAW-TEMP1
          IF(TEM1.GT.1.0/57.296) YAW=TEMP1+1.0/57.296
          IF(TEM1.LT.-1.0/57.296) YAW=TEMP1-1.0/57.296
          TEMP1=YAW
C
C  ERROR SIGNAL TO DRIVE RUDDER(YAW ACTUAL - YAW ORDERED)
          YAWE=YAW - YAWC
C
          S=((U*U)+(V*V))**.5
```

```
          DU=U-U1
C***IF D TYPE AUTOMATIC OPTIMAL CONTROLLER IS USED,*****
C***USE THESE                                      *****
      DS1=(YAWE - S1)/P1
      D=(S1 + DS1*Z1)*K+S2
C
C***IF OPTIMAL OR WELL TUNED PID IS USED TO GET GRAPHS***
C***            USE THIS PART.                         ***
C     PIDDV2=(YAWE - PIDV2)/ PIDT2
C     PIDDV3= (PIDV2- PIDV3 ) / PIDT2
C     D = (PIDK1*PIDK2*PIDT1*PIDDV3) + (PIDK1*PIDK2*YAWE)
C     C    + PIDV4
C
C***WHEN FINDING OPTIMAL PID PARAMETERS, USE THIS PART.****
C     PIDDV2=(YAWE - PIDV2)/ TT2(IT2)
C     PIDDV3= (PIDV2- PIDV3 ) / TT2(IT2)
C     D = (KK1(1)*KK2(IK2)*TT1(IT1)*PIDDV3) + PIDV4
C     C          +(KK1(1)*KK2(IK2)*YAWE)
C
C***  THE RUDDER ANGLE COULD NOT BE CHANGED MORE THAN  ****
C***  2.5 DEGREES PER  SECOND.                         ****
      TEM=D-TEMP
      IF(TEM.GT.2.5/57.296) D=TEMP+2.5/57.296
      IF(TEM.LT.-2.5/57.296) D=TEMP-2.5/57.296
      IF ( D.GT.(35./57.296) )   D =  35./57.296
      IF ( D.LT.(-35./57.296) )  D = -35./57.296
      TEMP=D
C
C*** THE VALUES OF HYDRODYNAMIC COEFFICIENTS TO REPRESENT**
C*** SHIP STEERING DYNAMICS. ( REF. 1, 6 )              **
C  AXIAL FORCE HYDRODYNAMIC COEFFICIENTS (SURGE)
      X1=(-0.00120)*(RO*L2*S)
      X2=(0.00045)*(RO*L2)
      X3=(-0.000103)*(RO*L2/S)
      X4=(-0.00898)*(RO*L2)
```

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```
              X5=(0.00018)*(RO*L4)
              X6=(-0.000948)*(RO*L2*S*S)
              X7=(0.00798)*(RO*L3)
              X8=(0.000832)*(RO*L2*S)
C     LATERAL FORCE HYDRODYNAMIC COEFFICIENTS (SWAY)
C         Y0=(-0.000036)*(RO*L2*S*S)
              Y0=0.0
              Y1=(-0.011604)*(RO*L2*S)
              Y2=(-0.08078)*(RO*L2/S)
              Y3=(-0.000038)*(RO*L2*S)
              Y4=(-0.00499)*(RO*L3*S)
              Y5=(0.15356)*(RO*L3/S)
              Y6=(0.002779)*(RO*L2*S*S)
              Y7=(-0.0009)*(RO*L2*S*S)
              Y8=(0.011896)*(RO*L2)
C     MOMENT ABOUT Z-AXIS HYDRODYNAMIC COEFFICIENTS (YAW)
C         N0=(0.000028)*(RO*L3*S*S)
              N0=0.0
              N1=(-0.002635)*(RO*L3*S)
              N2=(0.016361)*(RO*L3/S)
              N3=(0.000125)*(RO*L3*S)
              N4=(-0.00166)*(RO*L4*S)
              N5=(-0.05483)*(RO*L4/S)
              N6=(-0.001388)*(RO*L3*S*S)
              N7=(0.00045)*(RO*L3*S*S)
              N8=(-0.00489)*(RO*L3)
C     COMMON COEFFICIENTS
              C1=(0.0084)*(RO*L3)
              C2=(0.01546)*(RO*L3)
              C3=(-0.000086)*(RO*L4)
              C4=(0.000829)*(RO*L5)
              C5=(-0.000227)*(RO*L4)
C
C
C         REGULAR WAVES
```

```fortran
      FX=WA*RX*DCOS(WE*TIME+TX)
      FY=WA*RY*DCOS(WE*TIME+TY)
      MZ=WA*RZ*DCOS(WE*TIME+TZ)
C
      IF(TIME.EQ.0.0)FX=0.0
      IF(DABS(FY).LT.0.00000001)FY=0.0
      IF(DABS(MZ).LT.0.00000001)MZ=0.0
C
C
C  EQUATIONS OF MOTION
      F1 = X1*DU + X2*DU*DU + X3*DU*DU*DU + X4*V*V + X5*R*R
     1    + X6*D*D + X7*V*R + X8*V*D + RES + FX - FXX
      F2 = Y0 + Y1*V + Y2*V*V*V + Y3*V*D*D +  Y4*R
     1    + Y5*R*V*V  + Y6*D + Y7*D*D*D + Y8*D*V*V
     1    + FY + FYY
      F3 = N0 + N1*V + N2*V*V*V + N3*V*D*D + N4*R
     1    + N5*R*V*V  + N6*D + N7*D*D*D + N8*D*V*V
     1    + MZ + MZZ
C
C
C*** ADDED RESISTANCE RELEVANT DUE STEERING AND TOTAL *****
C*** RESISTANCE AFTER THE TRANSIENT TIME.             *****
C*** WHEN FINDING OPTIMAL PID PARAMETERS,             *****
C*** USE (TIME.LE.500.)                               *****
      IF (TIME.LE.1000.) GO TO 0001
      DELST1= ((X4*V*V+ X5*R*R+ X6*D*D+ X7*V*R+ X8*V*D)
     1              + DELST1)
      TOTFR2= (X4*V*V + X5*R*R +X1*DU +X2*DU*DU
     1           +X3*DU*DU*DU + X6*D*D + X7*V*R + X8*V*D
     1            + FX - FXX- 63420.) + TOTFR2
C
C
C*** FINDING ACCELERATION VALUES OF U, V, AND R  *****
0001  UDOT = F1/C1
      VDOT = (C4*F2-C3*F3)/(C2*C4-C5*C3)
```

```
                      RDOT = (C2*F3-C5*F2)/(C2*C4-C5*C3)
      C
      C
      C***   WHEN TO PRINTOUT                                    *****
             IF (ICOUNT.EQ.2) GO TO 50
             GO TO 300
      C
      C***   CONVERT RADIANS TO DEGREES                          *****
        50   YAWDEG= YAW*57.296
             RDEG=R*57.296
             RDDEG=RDOT*57.296
             DDEG=D*57.296
             YAWC=YAWC*57.296
      C
      C***   ONLY USE WHEN GETTING GRAPHS                        *******
             MH(ITIME+1)=DDEG
             T(ITIME+1)=TIME
      C
             ITIME=ITIME+1
             ICOUNT=1
      C
      C***   TEST IF WANT TO STOP                                *******
        300  IF (TIME.GT.ETIME) GO TO 400
      C
      C   INTEGRATION STEP SIZE DELT
             DELT=1.
      C
      C   INTEGRATION
             U=U+UDOT*DELT
             V=V+VDOT*DELT
             R=R+RDOT*DELT
             YAW=YAW+R*DELT
      C
      C   FOLLOWINGS ARE FOR D TYPE CONTROLLER.
             S1=S1+DS1*DELT
```

```
          S2=S2+(YAWE/KD)*DELT
C
C   FOLLOWINGS ARE FOR PID.
C       PIDV2=PIDV2+PIDDV2*DELT
C       PIDV3=PIDV3+PIDDV3*DELT
C       PIDV4=PIDV4+(YAWE*(PIDK1/PIDT3))*DELT
C
C*** SO MUCH SPEED DECREASING MAKE THE NONLINEAR*****
C*** COEFFICIENT VALUES INCORRECT TO REPRESENT   *****
C*** THE SHIP STEERING  DYNAMICS.                *****
C*** SO RPM (PROPELLER ROTATION PER MINUTE) IS   *****
C*** INCREASED TO PREVENT SHIP SPEED DECREASE.   *****
        IF(RPM.GT.95.) GO TO 65
        IF(U.LT.14.7*1.689) RPM=RPM+1./10
65      IF(U.GT.15.3*1.689) RPM=RPM-1./10.
C
C*** TO FIND SHIP MOTIONS ON THE EARTH SURFACE.  *****
C*** (REFS. 1, 9)                                *****
        XDOT = U*DCOS(YAW) - V*DSIN(YAW)
        YDOT = U*DSIN(YAW) + V*DCOS(YAW)
        X = X + XDOT*DELT
        Y = Y + YDOT*DELT
C
C***HOW MUCH ADDED FORCES OBTAINED BY INCREASING RPM*****
        RDIF=4480.*RPM-243908.
        RES=RDIF-63420.
C*** WITH USING SHIP ENGINE SPECIFICATIONS, THE FUEL    ***
C*** CONSUMPTION CAN BE FOUND.                          ***
        EHP=263.448*RPM-15153.5312
        IF ( RPM.LE.80. )    SHP = 250*RPM - 11900.
        IF ( RPM.GT.80. )    SHP = 420*RPM - 25500.
        BARFUE = BARFUE + (0.03125*SHP + 95.) * (DELT/86400.)
        LBFUEL = LBFUEL +(0.000007187*SHP + 0.632625)*SHP
     C                                          *(DELT/3600.)
C
```

```
          TIME=TIME+DELT
          ICOUNT=ICOUNT+1'
C***   FINDING APPROXIMATE COST FUNCTION          *****
          IF(TIME.GT.1000.)ISE=ISE + LAMDA*YAWE**2
          IF(TIME.GT.1000.)ISR=ISR + D**2
          TDIFF=(ISE + ISR)
C
          GO TO 200
    400 CONTINUE
C
C***   FINDING FUEL CONSUMPTION RATIO             *****
          DELST1 = DELST1 / (TIME - 1000.)
          TOTFR2 = TOTFR2 / (TIME - 1000.)
          PERC2  = (DELST1/TOTFR2) * 100.
C
          WRITE(6,1111) TDIFF,DELST1,PERC2
1111    FORMAT(1X,F7.3,1X,F8.0,1X,F8.3)
C
C***USE THIS PART TO PRINT ALL COMBINATIONS OF PID*****
C***PARAMETERS                                   *****
C       WRITE(8,43)KK1(1),KK2(IK2),TT1(IT1),TT2(IT2)
C    C              ,TT3(IT3),TDIFF
C       WRITE(6,43)KK1(1),KK2(IK2),TT1(IT1),TT2(IT2)
C    C              ,TT3(IT3),TDIFF
43      FORMAT(' ',5F8.2,2X,F20.5)
C
C***   DISSPLA STATEMENTS USING TO GET GRAPHS      *****
          XMIN=0
          XMAX=ETIME
          XINC=XMAX/6.
          MAGMAX=-1.E15
          MAGMIN= 1.E15
          DO 150 I=1,ITIME
          IF (MH(I).GT.MAGMAX) MAGMAX=MH(I)
150       IF (MH(I).LT.MAGMIN) MAGMIN=MH(I)
```

```fortran
      CALL AREA2D(6.0,3.0)
      CALL XNAME('TIME (#SEC.&) $',100)
      CALL YNAME('RUDDER (#DEG.&) $',100)
      CALL HEADIN('    $',100,1.2,4)
      CALL HEADIN('          SEA STATE=8     $',100,1.2,4)
      CALL HEADIN(' EN. ANGLE=030 SPEED=15
C                D TYPE CONT. $',100,1.2,4)
      CALL HEADIN('         WF = .53 ( EF =.33)
C                          $',100,1.2,4)
      CALL CROSS
      CALL YINTAX
      CALL GRAF(XMIN,XINC,XMAX,MAGMIN,'SCALE',MAGMAX)
      CALL CURVE(T,MH,ITIME,0)
      CALL RESET('CROSS')
      CALL ENDPL(0)
      CALL DONEPL
C
C*** USE THIS PART TO FIND OPTIMAL PID PARAMETERS     *****
C      IF (TDIFF.LT.SMALL ) THEN
C           SMALL=TDIFF
C      END IF
C55    CONTINUE
C44    CONTINUE
C33    CONTINUE
C22    CONTINUE
C11    CONTINUE
C      WRITE(8,*) SMALL
C
C

      STOP
      END
```

APPENDIX D
LIST OF SYMBOLS

ψ_c Commanded Heading Angle

ψ Yaw Angle, measured from the vertical xz plane to the axes of the ship; positive in the positive sense of rotation abiut the z-axes

ψ_e $\psi - \psi_c$

X   Hydrodynamic force components on ship body (longitudinal)

Y   Hydrodynamic force components on ship body (lateral)

N   Resultant total moments acting on a ship about the z-axis, yawing moment

u   Velocity components of the origin of the body axes relative to the fluid, longitudinal component

u_1 Commanded longitudinal velocity component

Δu $u - u_1$

v   Transverse component of velocity

r   Angular velocity of yaw

$\dot{u}, \dot{v}, \dot{r}$ Acceleration components of the origin of the the body axes relative to the fluid (longitudinal, transverse, and yawing, respectively)

δ Angular displacement of the rudder, measured from the xz-plane of the ship to the plane of the rudder

m   Mass of ship

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X_G	Coordinate of the center of mass of the ship ship relative to body axes
X^*, Y^*, N^*	Values of X, Y, and N at $v=r=\dot{v}=\dot{r}=0$ and $u=u_1$
W_a	Significant wave height
R_i	Exciting force magnitude
ϕ_i	Exciting force phase angle
W_e	Encounter frequency
W_{wave}	Wave circular frequency
β	Encounter angle
g	Acceleration of gravity
LBP	Length between perpendiculars
S	$(u^2 + v^2)^{1/2}$
w	Natural frequency of ship's steering
OP	Distance from the ship pivot point to the origin (0.3 L)
L	Ship length
λ	Weighting factor
r_a, v_a, δ_a	Amplitude of r, v, and δ
P	Engine power
W_{net}	Work done per unit mass (btu / lbm)
m	Fuel mass per unit time (lbm / min)
X_{calm}	R = T (1 - t), total ship resistance without propeller in calm water case
t	Thrust deduction fraction
T	Propeller thrust

X_{vv} Partial derivative of X with respect to v^2

Y_{rvv} Partial derivative of Y with respect to rv^2

N_{vvv} Partial derivative of N with respect to v^3

x_0, y_0, z_0 System of reference axes through origin
      of reference axes fixed in the ship
      whose direction fixed in the ship

x, y, z System of reference axes whose origin
      and direction remain fixed in the earth

Notes:

1. Signs of all directions, forces, distances, velocities, and accelerations are positive downward along the z-axes, positive to starboard along the y-axes, and positive forward along the x-axes and similarly along the x_0, y_0, and z_0-axes

2. Signs of all angles, angular velocities, angular accelerations, and moments are positive if clockwise when facing in the positive direction of appropriate axes.

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